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PNEUMATIC NUTATOR ACTUATOR MOTOR

By

C. N. High, G. R. Howland, J. R. Williamson

Prepared For

NATIONAL AERONAUTICS & SPACE ADMINSTRATION

CONTRACT NAS3-5214

THE Bendix CORPORATION

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FIRST QUARTERLY REPORT

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ABSTRACT

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This is a first quarter report of a twelve-month program to design, fabricate and test a prototype pneumatic nutator actuator motor for drum control of a nuclear reactor.

The high torque, low speed motor contains an integral transmission consisting of a pair of bevel gears, one of which is operated in a nutation motion by a number of pressurized bellows located around its periphery. Pneumatic flow is commutated to the power bellows by means of a number of vortex type fluid interaction devices.

During this reporting period both mechanical and fluid state element designs were established. Tests were performed on components of the commutation circuit, the results of which are described in the report. Based upon these tests, the dimensions of the power and selector vortex valves which form a major portion of the commutation circuiting were established. Fabrication status of all components is described.

Author

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SUMMARY

This report describes the concept and first quarter accomplishments of a twelve-month contract to develop a pneumatic actuator motor of a new concept. The actuator motor operates from a pneumatic supply and produces a high torque, low speed mechanical output proportional to a pneumatic input pressure differential signal. The commutation logic of the motor is accomplished by closed loop fluid interaction (vortex type) devices.

The mechanical design of the motor is described as well as the logic commutation circuit. Test results are given of critical logic circuit components to demonstrate circuit feasibility. Discussions of vortex valve optimization studies and transfer plate sealing tests are also given.

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SECTION 1

ACTUATOR CONCEPT

1.1 INTRODUCTION

The Pneumatic Nutator Actuator Motor is presently being developed by Bendix Products Aerospace Division under Contract NAS3-5214 for NASA-Lewis Research Center. The purpose of the contract is to build and evaluate a pneumatic actuator motor for control of a nuclear reactor. The required performance is given in the specifications of Contract NAS3-5214. The motor is of a different concept from the conventional gear, vane or piston type actuators. The logic circuits required to operate the motor have no moving parts and are composed of fluid interaction (vortex type) devices.

1.2 NUTATOR PRINCIPLE

In order to provide a low speed, high torque output, the actuator motor makes use of the nutating gear concept to obtain a high gear reduction. The nutator transmission consists of a pair of bevel gears. One gear is attached to the output shaft, and the other is attached to the case through a pair of gimbal rings. The gimbal rings allow the input gear to wobble or nutate, but not rotate.

If a force is applied at a point on the circumference of the input gear, the gears will mesh along the pitch line of a single tooth. By moving the point of application of the force around the circumference, the input gear will mesh consecutively with each of the teeth in the output gear. If the output gear has one more (or less) tooth than the input gear, the output will be displaced by one tooth for each complete nutation of the input gear. Gear reductions between 50:1 to 200:1 can be accomplished in this manner by a single pair of gears.

1.3 ACTUATOR PRINCIPLE

The nutator motor actuator employs the above described principle. The force can be applied to the input gear by a number of electromagnetic solenoids (for an electrical input) or by pressure force elements (bellows or pistons) for a hydraulic or pneumatic input.

The pneumatic nutator actuator motor being developed under this contract employs eight bellows assemblies. Four of the bellows are pressurized at any given time. Nutation is caused by increasing the pressure to the fifth bellows and simultaneously decreasing the pressure in the first bellows. The advantages of using bellows are given in Section 2.

The actuator can be employed as an analog device or as a digital stepping motor depending on the type of commutation used. The commutation being developed is of the analog type and is described in Subsection 1.5.

A schematic of the actuator motor is shown in Figure 1-3-1.

1.4 ADVANTAGES OF THE NUTATOR ACTUATOR

The nutator actuator has a number of desirable features which make it superior to a conventional actuator, particularly where extreme environmental conditions exist. These advantages may be outlined as follows:

(a) Minimum number of moving parts. The only rotary motion is in the slow moving output gear and shaft. Only two bearings are employed in the entire actuator. The input motion is a nutation motion consisting of a small (±.060 inch) axial movement.

(b) High speed of response. Since there are no high speed rotating components, the input inertia is low. This allows a rapid reversal of direction to a step command. The high response has been demonstrated on a prototype electromechanical nutator actuator.

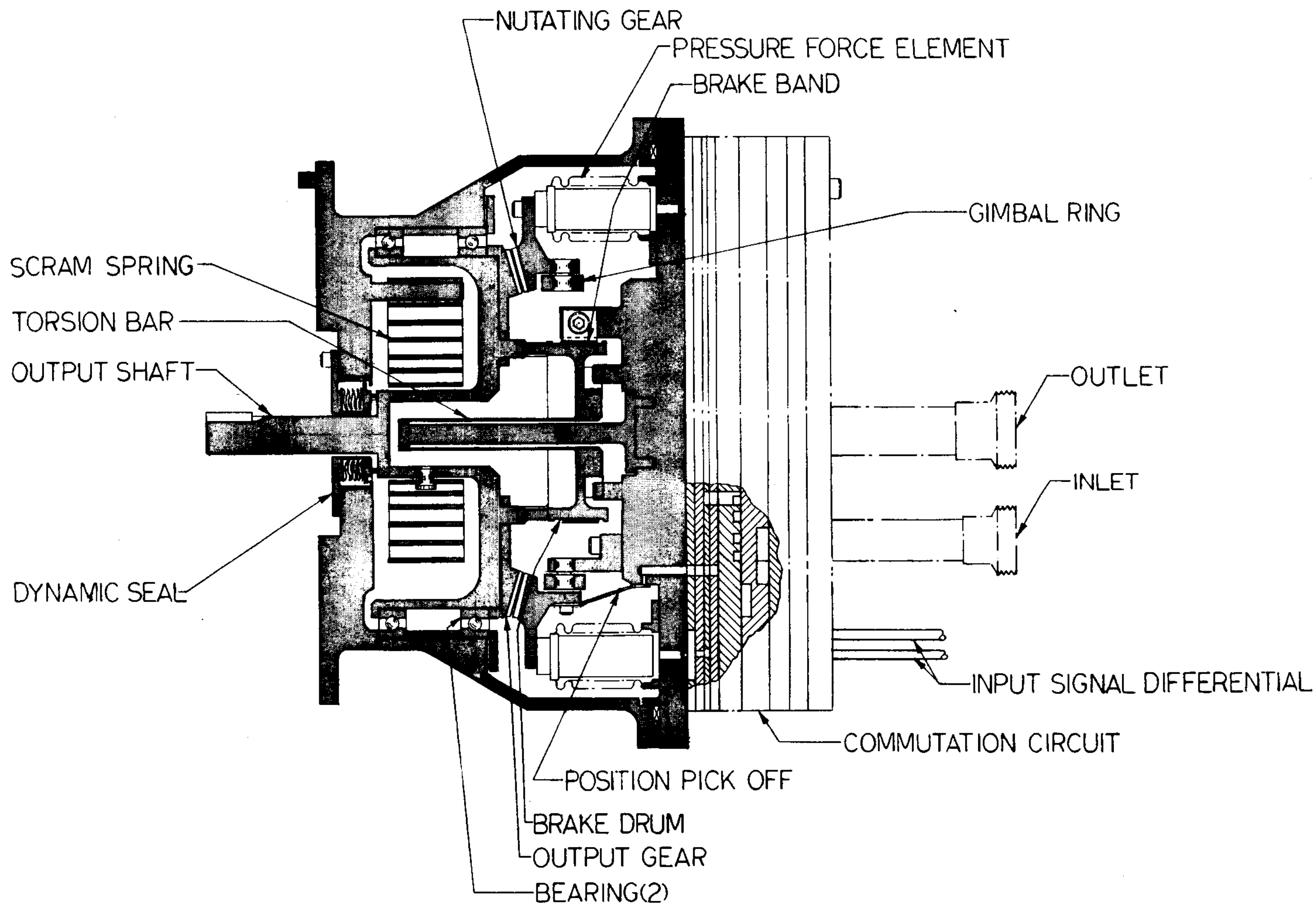


FIGURE 1-3-1 PNEUMATIC NUTATING ACTUATOR MOTOR

(c) Automatic declutch. If the pressure to the bellows is nulled, the spring rate of the bellows will cause the nutating gear to disengage and lie in a plane parallel to the output gear. The output shaft is then free of the input. A spiral preloaded spring will then drive the output shaft to the zero angle position. A small brake drum and torque tube absorb the energy of impact. This scram feature, which is mandatory in nuclear reactor control, is inherent in the nutator actuator design.

(d) Fail safe features. The most critical component in the actuator is the pressure force elements. A bellows failure will not cause a seizure but only a reduction in output torque. Multiple bellows failure will result in disengagement of the gears, and a scram to zero will occur.

1.5 METHOD OF COMMUTATION

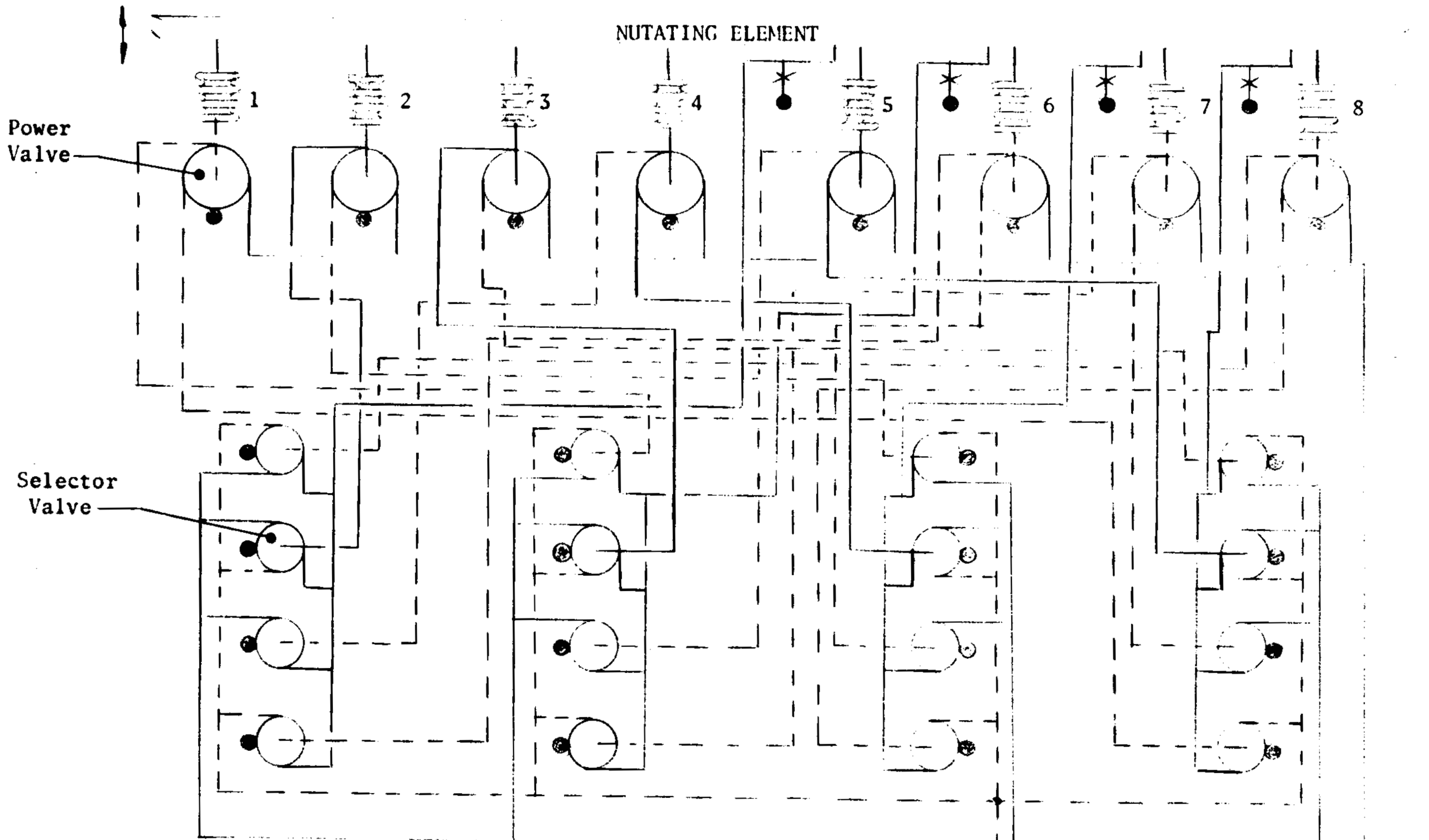
The actuator motor can be operated as a digital stepping motor if the appropriate commutation circuit is employed. For the actuator motor under design, a pulse input would cause an output step of 0.25 degree.

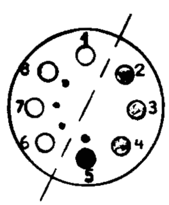
The commutation circuit for the preliminary model will be a proportional closed loop analog design. The input is a pressure differential which results in an output torque proportional in magnitude and direction to the input differential.

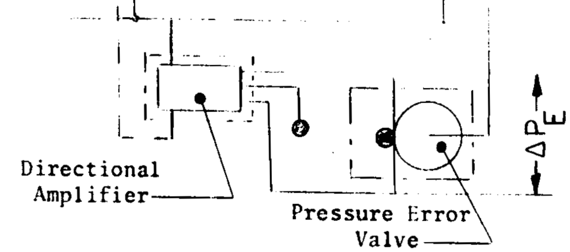
The position of the input nutating gear is indicated by a leaf spring attached to the gear which covers a bleed hole in the actuator motor case (see schematic). Four bleed holes are used, equally spaced around one-half of the circumference. Each bleed hole is a downstream variable restrictor of two restrictors in series. The intermediate pressures obtained are therefore an indication of the input gear position.

Four of the eight bellows are pressurized at any given time. The four bellows pressurized will cause the nutating gear to assume a given position. The resultant change in pickoff pressures will cause the fifth bellows to become pressurized and the first bellows to reduce in pressure. As the gear moves under the pressure forces, the pickoff pressures will again change, resulting in a further change in bellows pressure. The commutation is therefore similar to a conventional electric motor, except that the action is proportional.

An example of the operation of the commutation circuit is illustrated in Figures 1-5-1 through 1-5-4.





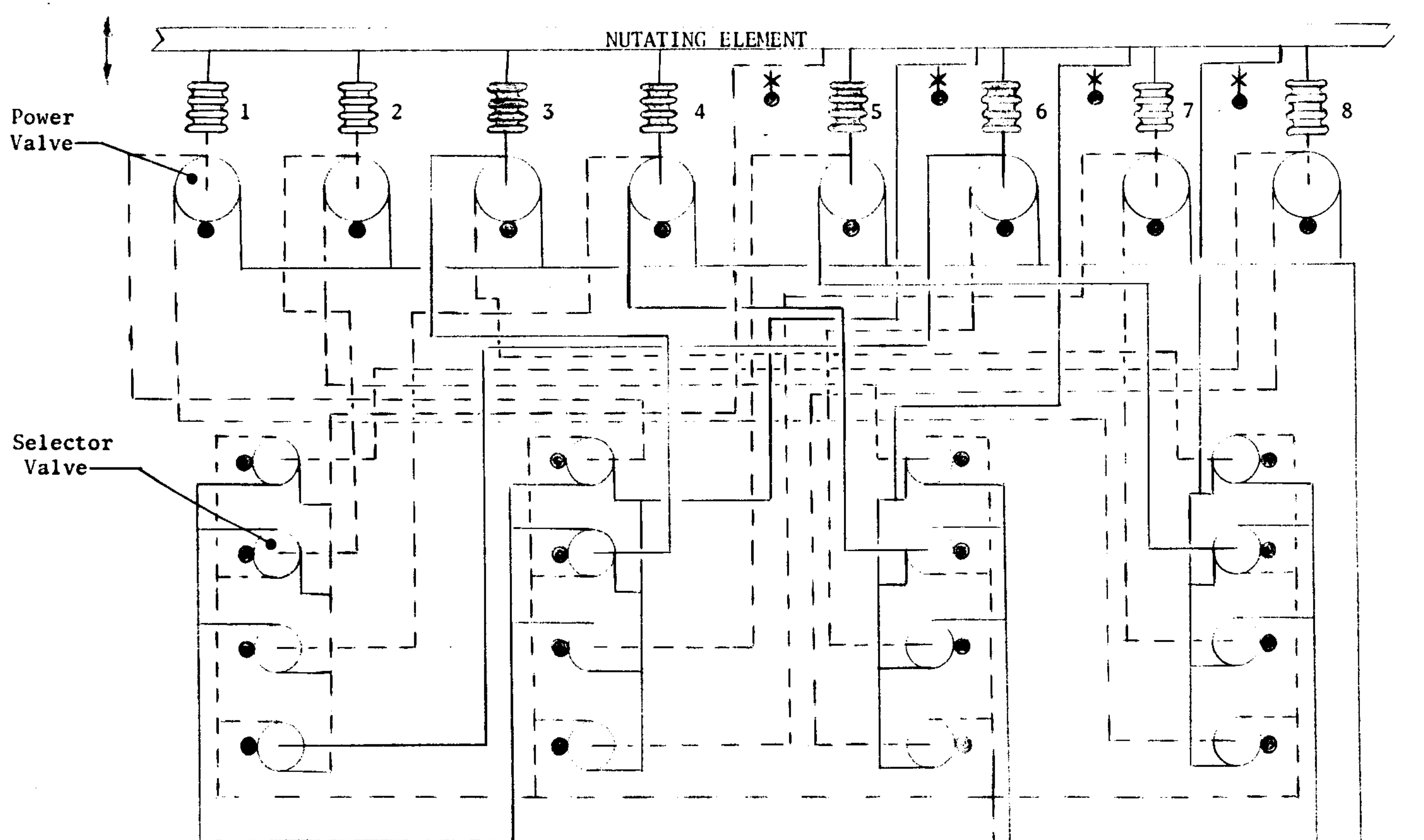


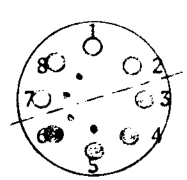
● - Denotes Supply Pressure
Solid Lines Indicate High Pressure

DESCRIPTION OF OPERATION

At the initiation of operation, the nutating elements are parallel and the position pickoffs' intermediate pressures are all above the threshold pressure. Signals are sent back from all pickoffs to the selector valves and bellows 2 through 5 are all pressurized.

FIGURE 1-5-1 NUTATING ELEMENT





◉ - Denotes Supply Pressure
Solid Lines Indicate High Pressure

DESCRIPTION OF OPERATION

As soon as the nutating element begins to lift, the intermediate pressure of pickoff S_1 decreases. When its value is below that of the signal threshold, the second bellows loses pressure and the sixth bellows begins to fill. This moves the neutral axis of the nutating element 45° clockwise.

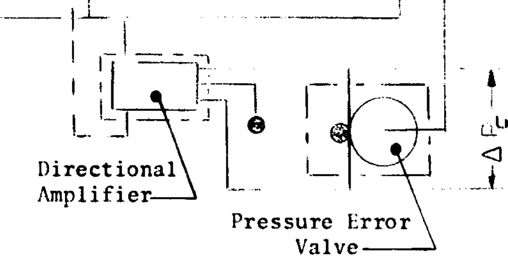
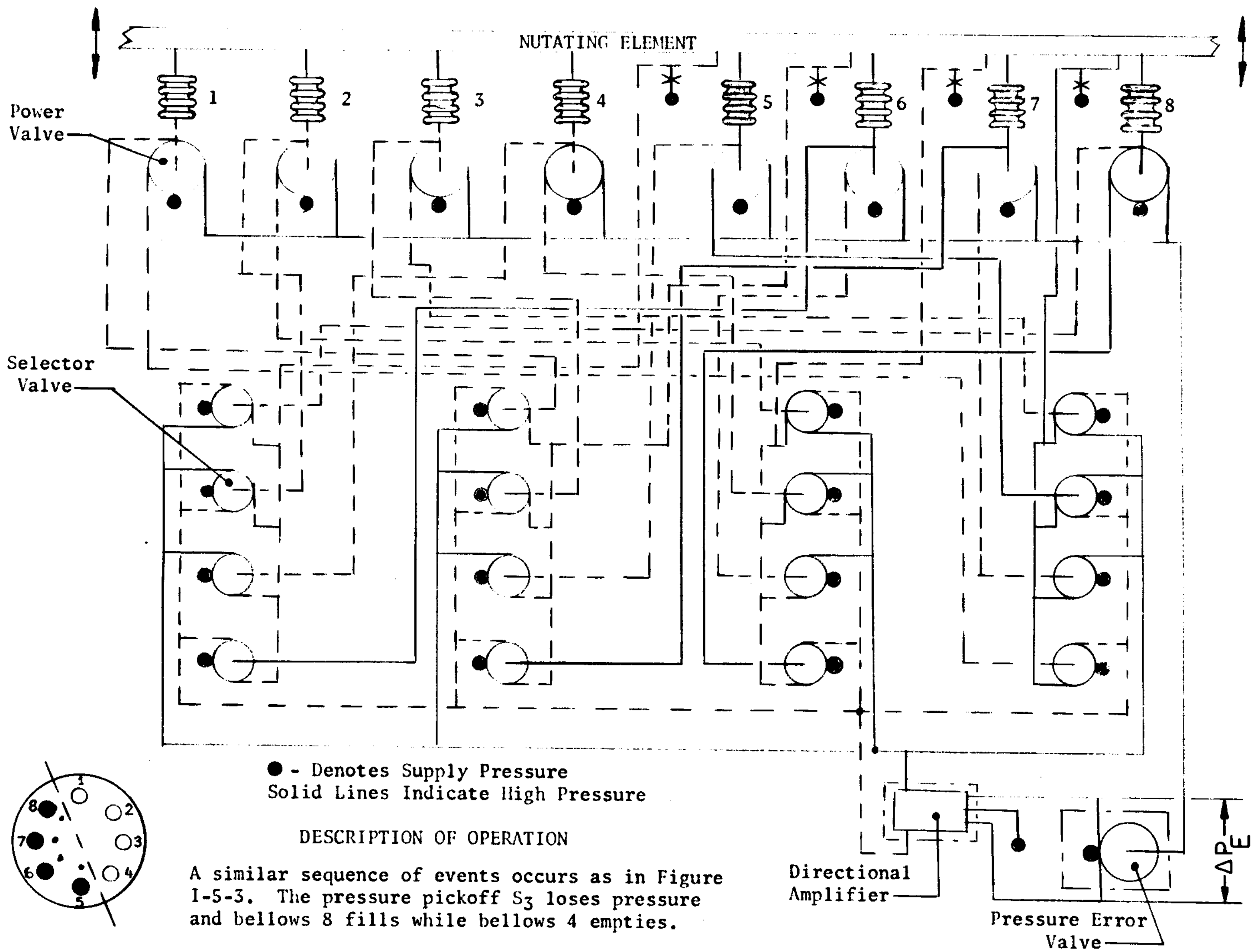


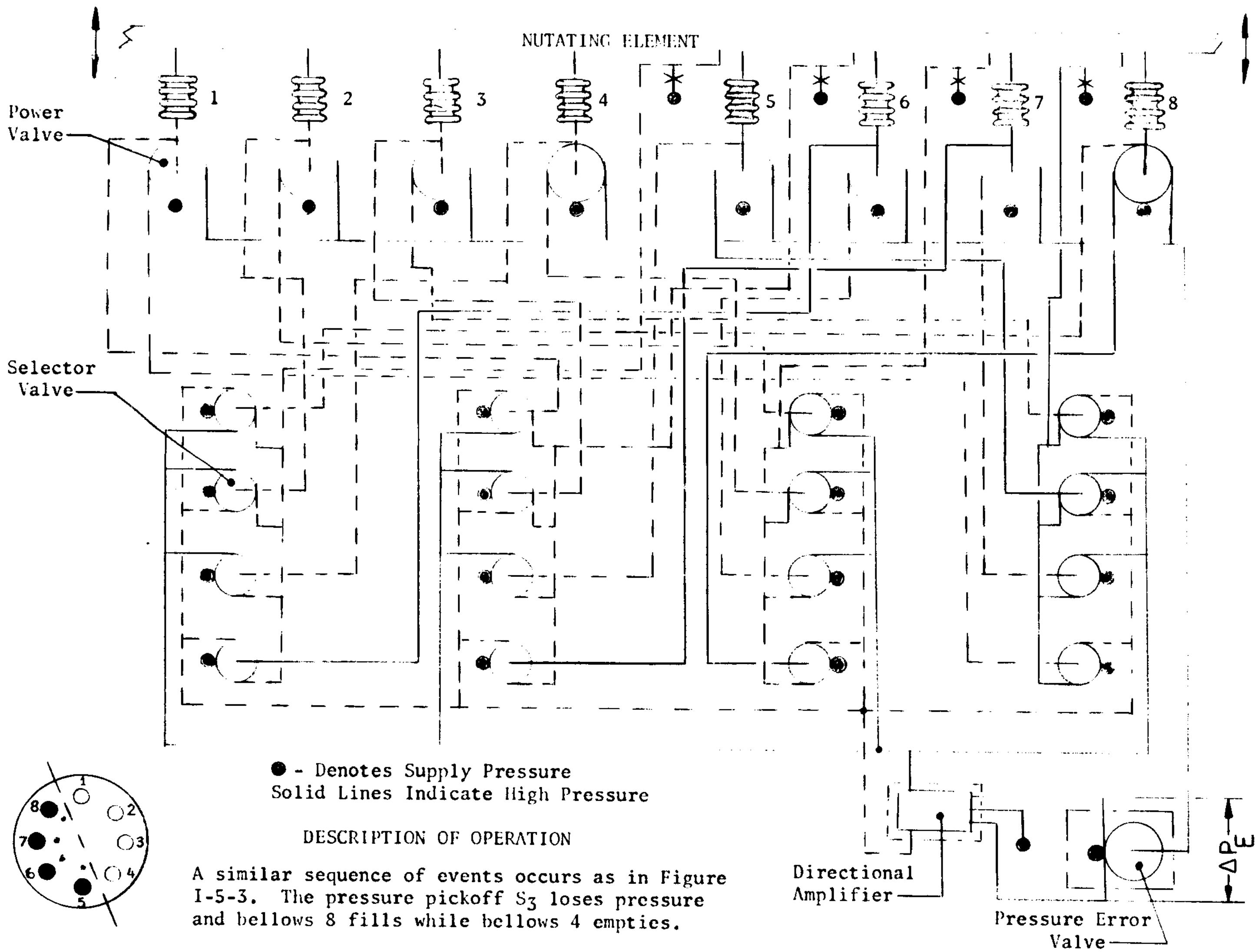
FIGURE 1-5-2 NUTATING ELEMENT



DESCRIPTION OF OPERATION

A similar sequence of events occurs as in Figure I-5-3. The pressure pickoff S_3 loses pressure and bellows 8 fills while bellows 4 empties.

FIGURE 1-5-3 NUTATING ELEMENT



DESCRIPTION OF OPERATION

A similar sequence of events occurs as in Figure I-5-3. The pressure pickoff S_3 loses pressure and bellows 8 fills while bellows 4 empties.

FIGURE 1-5-4 NUTATING ELEMENT

SECTION 2

MECHANICAL DESIGN

2.1 COMPONENTS

2.1.1 Gears

The gears designed for this application are based on principles used to design gears for nutator transmissions manufactured by Bendix over the past four years. They are a straight-flank bevel design and, in this case, have a cone angle of 140 degrees. The input gear has 181 teeth and the output gear 180 teeth, resulting in a reduction ratio of 180:1. Both gears are being made of AMS 5643 heat treated to Rockwell C 35-42. The teeth are to be treated with 0.0003-0.0007 inch thick coating of Hi-T-Lube, which is a dry lubrication process applied by the General Magnaplate Corporation. Of the many nutator transmissions built to date, all gears and bearings have used this lubrication. No gear or bearing failure has occurred during normal operational testing and some customer units have accumulated over 200 hours of operation.

To permit its nutation, the input gear is supported by a gimbal mounting incorporating four Bendix "Free-Flex" frictionless pivots. This arrangement provides axial rigidity while allowing the input gear to nutate freely with no "play" or waste motion at the pivot points.

2.1.2 Bearings

For the past six years, Bendix has supported an intensive test program to develop bearing materials which have a long operating life in extreme environments. Tests conducted by the Research Laboratories and Products Aerospace Divisions have utilized a Bendix' designed, rolling contact bearing test machine which has an ambient temperature capability of 100°R to 2640°R and a pressure range from 10^{-9} mm Hg to 400 psia with a variety of ambient gases. In addition, various dry-film lubricants for use in bearings have been tested in cryogenic environments resulting in the selection of two similar lubrication methods which produce excellent results.

With the first method, the bearing balls and races are made of Type 440C stainless steel. The ribbon-type ball separators are made of Type 300 series stainless steel and coated with Hi-T-Lube. In operation, the balls transfer the dry film material from the separators to the races to provide a lubricating film.

The second method again uses bearings containing Type 440C stainless steel balls and races with ball separators made of duPont's SP-1 Polymer. Bearing action transfers the SP-1 material to the metal parts of the bearing to provide a lubricating film. Comparative tests indicate that in a hydrogen atmosphere this method is slightly superior to the Hi-T-Lube method with respect to bearing life. However, either lubricating method will be satisfactory and result in a device which can meet specification requirements. Choice of material depends to some extent upon bearing size and ease of manufacture of the ball separators.

Pertaining to the bearings for the actuator motor, only two ball bearings will be used. Angular contact type bearings have been selected. These bearings axially locate the output shaft, absorb the thrust load caused by the gear separating force and absorb the pressure force created by the dynamic shaft seal. The selection of bearing size was based on the maximum bearing load, the mean effective speed, and a B-10 catalog life based on at least ten times the desired life of the actuator. Experience has shown that this method of derating provides satisfactory results.

2.1.3 Scram Mechanism

A scram mechanism is required to return the output shaft to the zero position on initiation of a scram command signal or the loss of pneumatic pressure.

The pneumatic pressure to the actuator may be interrupted due to a failure or by a scram command signal. This interruption automatically disengages the transmission and the load inertia becomes free-wheeling. A spiral spring preloaded to the output shaft ensures the required scram action. A schematic drawing of the scram system is shown in Figure 2-1-1 and a summary of scram characteristics based on a constant dynamic load friction of 32 inch-pounds is shown in the table below. This scram concept has been used by Bendix on previous projects with environments of nitrogen and hydrogen gas and a temperature range from -260°F to +650°F. The spring material and operating stress level will be based upon this past experience. Bendix Engineering Specification CNPD-188, which covers the scram spring, is included in Appendix A.

SCRAM CHARACTERISTICS

Maximum Scram Time	0.21 second
Maximum Impact Velocity	21.7 radian/second
Spring Size (Approximately)	0.052 x 1.25 cross section (11 turns)
Spiral Spring Rate	4 in. lbs./radian
Preload	50 in. lbs. (at zero)

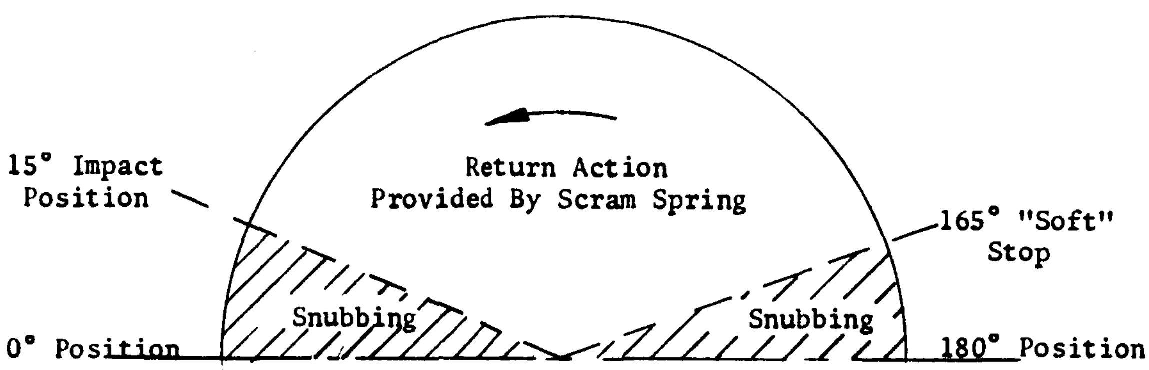


FIGURE 2-1-1 SCRAM SYSTEM - SCHEMATIC

2.1.4 Snubbing Mechanism

A snubbing device is required to ensure that the deceleration on completion of scram action is less than 2000 radians/second. The snubbing action is designed to take place during the 15-degree to zero-degree portion of output shaft travel.

A lever, keyed to the output shaft, engages a small brake drum, which is attached to the actuator housing with a torsion bar. The mechanism is designed so that the torsion bar will store a portion of the kinetic energy so that the output shaft is capable of returning to the 15-degree position under all conditions while the brake will dissipate a portion of the kinetic energy sufficient to prevent excessive rebound.

SNUBBER DESIGN DATA

Impact Velocity	2.17 radians/second
Kinetic Energy at Impact (Maximum)*	91 inch-pounds
Kinetic Energy Absorbed by Brake (Maximum)*	37 inch-pounds
Kinetic Energy Absorbed by Torsion Bar (Maximum)*	54 inch-pounds

* Based on a release position at 150° from impact with brake.

In addition to the stop at the 15-degree position, the brake is also designed to engage at the 165-degree position to prevent actuator damage due to an inadvertent overtravel command.

2.1.5 Dynamic Shaft Seal

The dynamic shaft seal selected for this application is a face-type seal utilizing a welded metal bellows to maintain contact between the face of the seal and a specially prepared surface on the output shaft.

In the design and construction of this type of seal, two goals are desired; a low leakage rate and a low operating torque. If sufficient bellows spring force is utilized to obtain near zero leakage, the torque required to overcome the friction will rise to an unacceptable level. In similar programs, a design goal of 0.001 lb./sec. of gaseous H_2 leakage and 15 inch-pounds of torque was established as the maximum acceptable limit for all specified gas temperatures and pressures.

The seal is installed into the actuator, as shown in the layout, so that the pressure load is supported by the flanged portion of the housing. Leakage around the seal case is prevented by using a static seal. The seal can be installed or removed from the actuator without disturbing other components. This arrangement greatly facilitates initial installation adjustments, test inspections, and seal replacement.

A typical bellows-type face seal is shown in Figure 2-1-2. This seal consists of a welded Inconel X bellows assembly, attached to a back bellows plate, and a front plate or cup containing a carbon seal ring.

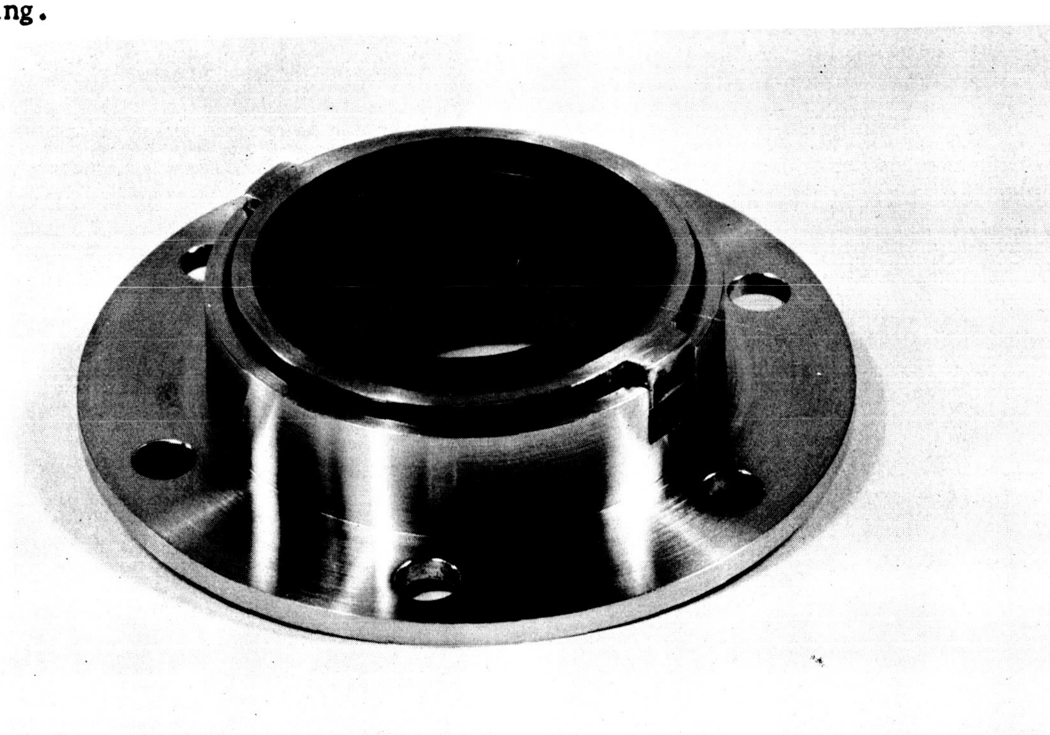


FIGURE 2-1-2 BELLOWS TYPE FACE SEAL

A seal material evaluation study was conducted at Bendix to determine the optimum dynamic sealing materials combinations for use in actuators exposed to nuclear-cryogenic environments. Forty-seven various seal materials were tested. The results of this test program are reflected in our present dynamic seal designs. The dynamic seal used in the latest model turbine power control valve (TPCV) actuator for the NERVA program performed successfully over a pressure range of -65 to +650 psid and through a temperature range of -280 to +1300°F for a total accumulated operating time in excess of 100 hours.

Bendix Engineering Specification CNPD-189, which covers the dynamic shaft seal, is included in Appendix B.

2.1.6 Pressure-Force Elements

One of the critical components of the nutator motor is the pressure-force element. This device must convert the pneumatic pressure into a force efficiently and rapidly. Hence, it must be a device having minimum leakage, minimum friction, low inertia, minimum displacement volume, and a very high cycle life.

The bellows unit, Figure 2-1-3, is composed of a metal bellows, two end fittings, and a thin-walled cylinder placed in the center of the unit to reduce its internal volume. The unit is an interchangeable subassembly.

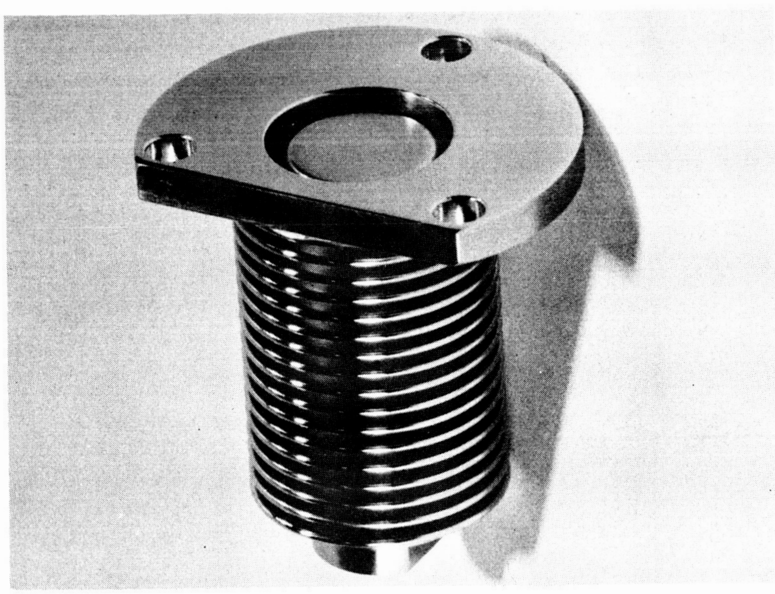


FIGURE 2-1-3 BELLOWS UNIT

The metal bellows is an electrodeposited type manufactured by the Servometer Corporation of New Jersey. It is made of material composed of 99.5% nickel, 0.4% cobalt, and traces of oxygen and carbon. Based on tests conducted by the manufacturer, it has an average cycle life of 50×10^6 cycles at -260°F for a stroke of 0.10 inch and a maximum pressure drop of 85 psi. Using a value of 40 degrees/second as a rated output velocity, a transmission ratio of 180:1 and considering eight pressure-force units, it is determined that each bellows must cycle at a rate of twenty cycles per second. Based on the cycle life given and the values assumed, the pressure-force elements would have an expected life of almost 700 hours. There are several advantages associated with the bellows unit such as dirt insensitivity, the elimination of all sliding or rubbing that is normally associated with a piston element, the no-leakage characteristic, the ability to accept some slight misalignment of the two end connections during operation, and the low mass of this thin-wall member.

Specification CNPD-187, covering the bellows assembly, is included in Appendix C.

2.2 PRESENT STATUS

2.2.1 Actuator Motor

The layout, assembly drawing, and detail drawings for the mechanical portion of the actuator have been completed, checked, and sent to Material Control for procurement. Specifications were written for the dynamic shaft seal, the scram spring, and the bellows assembly.

2.2.2 Test Adaptor

An adaptor was designed and released for procurement. This adaptor will permit testing the actuator on existing drum control fixtures. Provision is made in the adaptor for the use of two position indicating potentiometers.

2.2.3 Mechanical Commutator

The purpose of the mechanical commutator is to permit testing the motor actuator independent of the pure fluid commutator. Functionally, the mechanical commutator is nothing more than a series of three-way valves which sequence the pressurizing and exhausting of the eight bellows units of the actuator.

The device is composed of a shaft driven, combination spool and plate valve and a manifolded mounting plate which attaches to the rear of the motor. Provisions have been made for the insertion of pressure pickups in the channels supplying the bellows units.

The mechanical commutator is drawn and detailed and the drawings are presently being checked.

SECTION 3

COMMUTATION CIRCUIT

3.1 DEVELOPMENT OF VORTEX VALVE COMMUTATION CIRCUIT

3.1.1 The Analog Self Commutation Circuit

The analog self commutation circuit is shown schematically in Figure 3-1-1. The circuit for four of the eight bellows is shown. The remaining circuits are identical.

The input pressure differential (ΔP_e) is applied across two opposing control ports of the pressure error vortex valve (A_1) as well as across the bistable directional unit (D_1). A vortex swirl is generated in A1 either clockwise or counterclockwise depending on the sign of (ΔP_e). The output pressure of vortex valve A_1 will therefore reduce as a linear function of the absolute value of ΔP_e and sets a bias on the power vortex valves B_1 to B_8. The output of the power valves B_1 to B_8 supply the actuator motor bellows with flow and pressure. The maximum pressure level obtainable in the bellows is inversely proportional to the bias control pressure and therefore is directly proportional to ΔP_e.

One output of the directional control bistable unit (D_1) forms one input to eight AND units. The second input to the AND units is one output of the corresponding commutator unit, C. The commutator units C_1 to C_4 are proportional gate valves with the control port vented. The four commutation units are equi-spaced on a half circle, facing the nutating gear. When the gears mesh near the position of the commutator unit, the nutating gear motion blocks the control port and causes the gate valve to shift outputs. The gate pressure outputs are therefore an indication of nutator position.

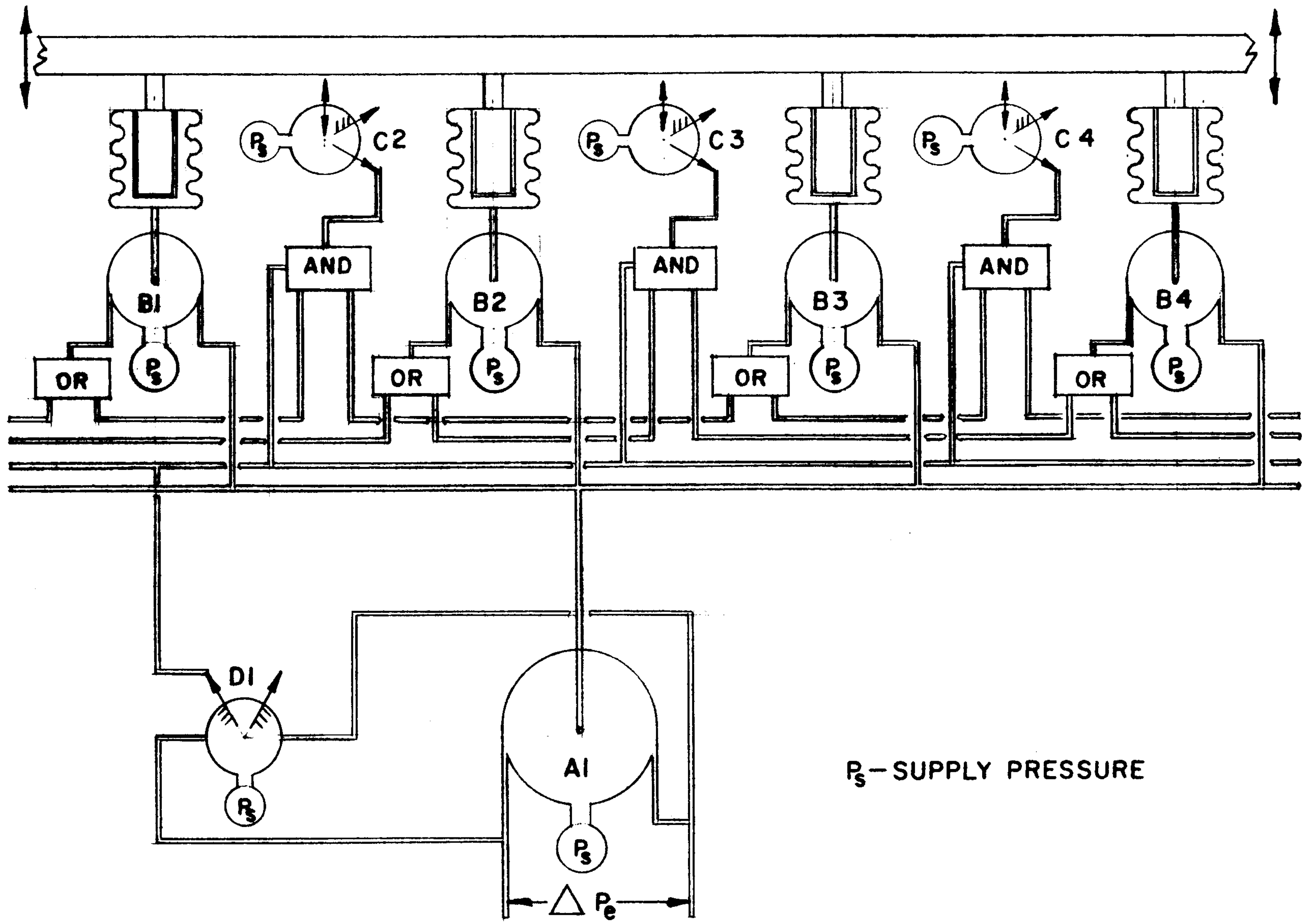


FIGURE 3-1-1 PARTIAL ANALOG SELF COMMUTATION CIRCUIT

When an output pressure occurs at C_2, pressure will occur at either the AND output or the AND-NOT output depending on the sign of the input error. This pressure opposes the bias flow in the corresponding power valve (B_3 or B_1), causing the pressure in the bellows to increase and forcing the nutating gear to move. The control port of C_3 (or C_1 depending on which bellows was pressurized) is then blocked and the sequence repeats.

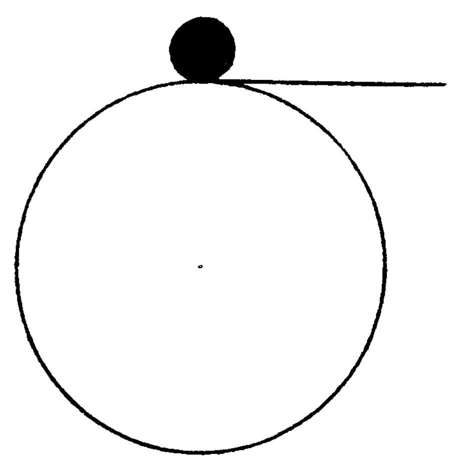
The minimum pressure in the activated bellows with no error input will be sufficiently high to hold the gears in mesh against the scram spring preload but will not cause rotation of the output shaft.

3.1.2 Development of the Vortex Valve

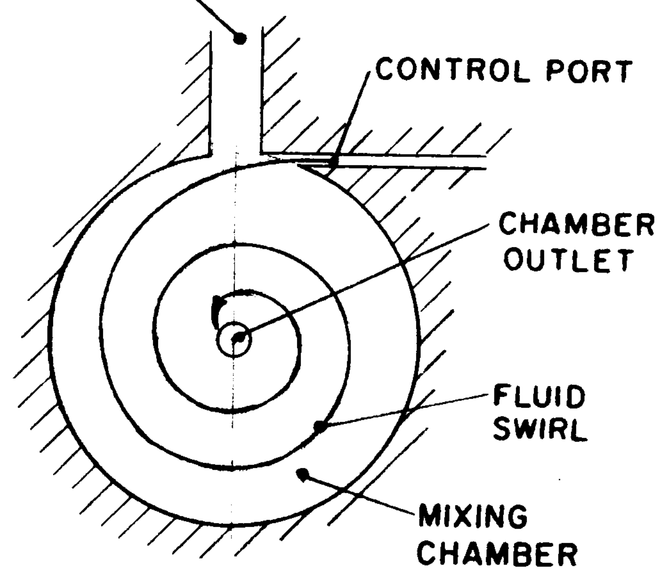
Figure 3-1-2 shows schematically the operation of a conventional vortex valve. The vortex valve consists of a cylindrical chamber in which supply flow is introduced radially at the periphery. The outlet is on the axis of symmetry. Control flow is injected tangentially at the periphery. The interaction of the control and main flows cause the combined flow to rotate in a spiral towards the outlet. As this spiral develops, the impedance to supply flow increases, reducing the supply flow. In the extreme condition of complete turndown, the supply flow is reduced to zero, and the outlet flow is equal to the control flow. Since the maximum control flow is less than the original supply flow, the vortex valve acts as a throttling device.

If a second outlet is provided in the vortex valve as shown in Figure 3-1-3, the pressure at this outlet is a direct indication of the total mass flow through the valve. Another important property of the second outlet is that when the valve is in complete turndown, pressure applied to the second outlet will cause a flow to occur across the vortex chamber and through the opposite outlet, with relatively low impedance. This is illustrated in Figure 3-1-4.

The vortex valve with two outlets can be used very advantageously as the power vortex valve. Figure 3-1-5 illustrates filling and emptying the bellows with the power vortex valve.



VORTEX VALVE — SYMBOL



VORTEX VALVE — SCHEMATIC

FIGURE 3-1-2 VORTEX VALVE - SCHEMATIC

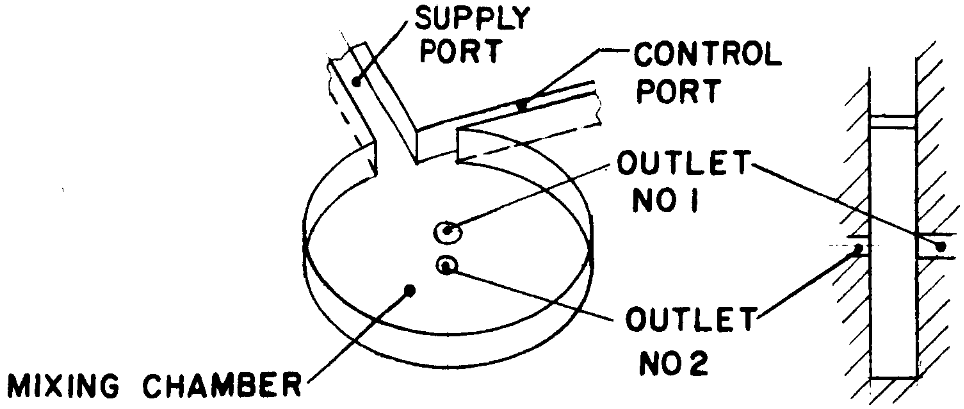


FIGURE 3-1-3 VORTEX VALVE WITH TWO OUTLETS - SCHEMATIC

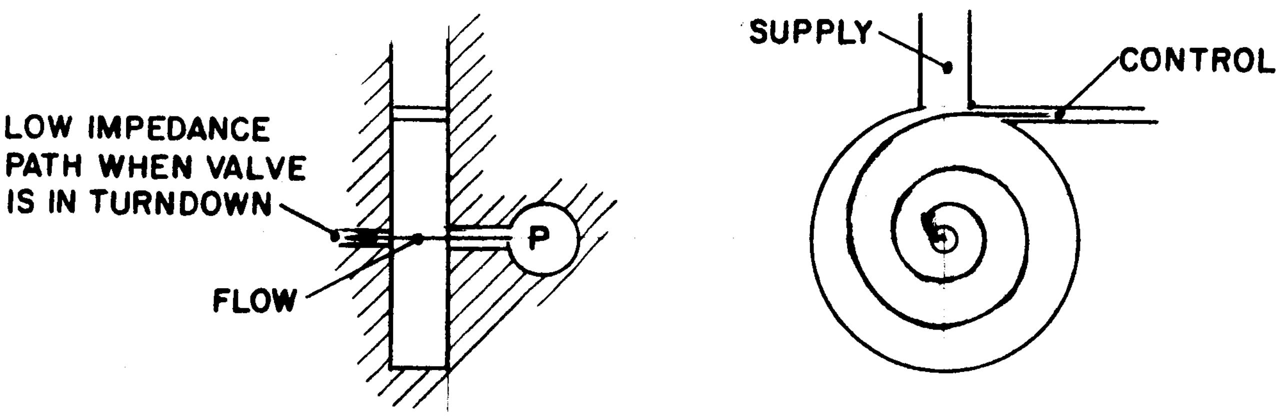


FIGURE 3-1-4 VORTEX VALVE WITH CROSSFLOW - SCHEMATIC

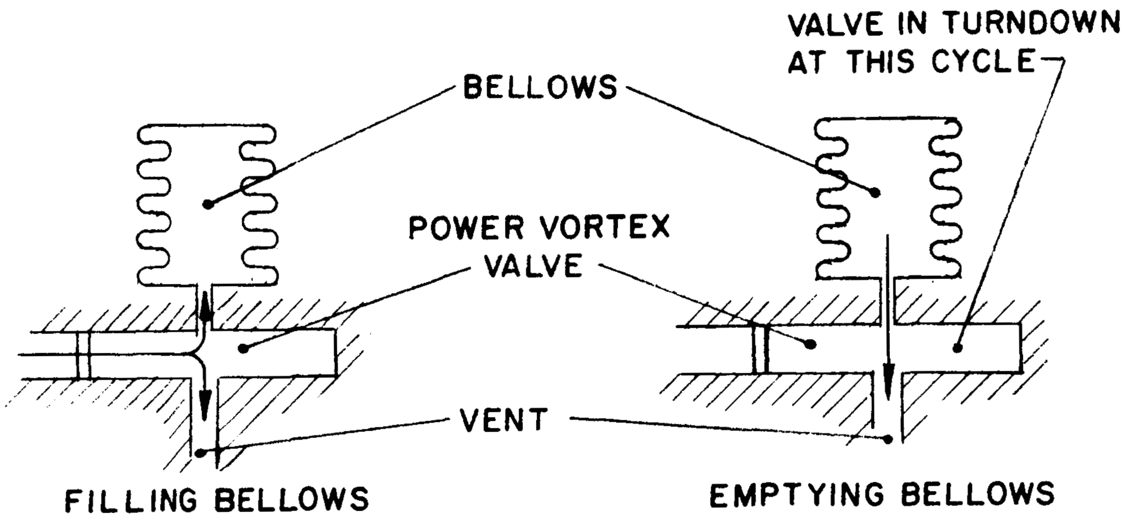


FIGURE 3-1-5 VORTEX VALVE-BELLOWS INTERACTION - SCHEMATIC

Algebraic addition can also be accomplished with vortex valves. Figure 3-1-6 illustrates a vortex valve with opposed control ports. If one control port is activated, the valve will be in turndown. If the opposite control port is then also activated, the output pressure will be inversely proportional to the difference between the control port pressures. Figure 3-1-7 illustrates a vortex valve with control pressures adding. In this case, the output pressure is inversely proportional to the sum of the control pressures.

Vortex valves can be employed to perform the same functions as the AND, OR, and gate valve components employed in the analog commutation circuit. Figure 3-1-8 illustrates vortex valves performing an AND function. Pressure signals are denoted by "X". Figure 3-1-9 illustrates the use of vortex valves as the gate valve function. Figure 3-1-10 illustrates the incorporation of the OR function directly into the power valve.

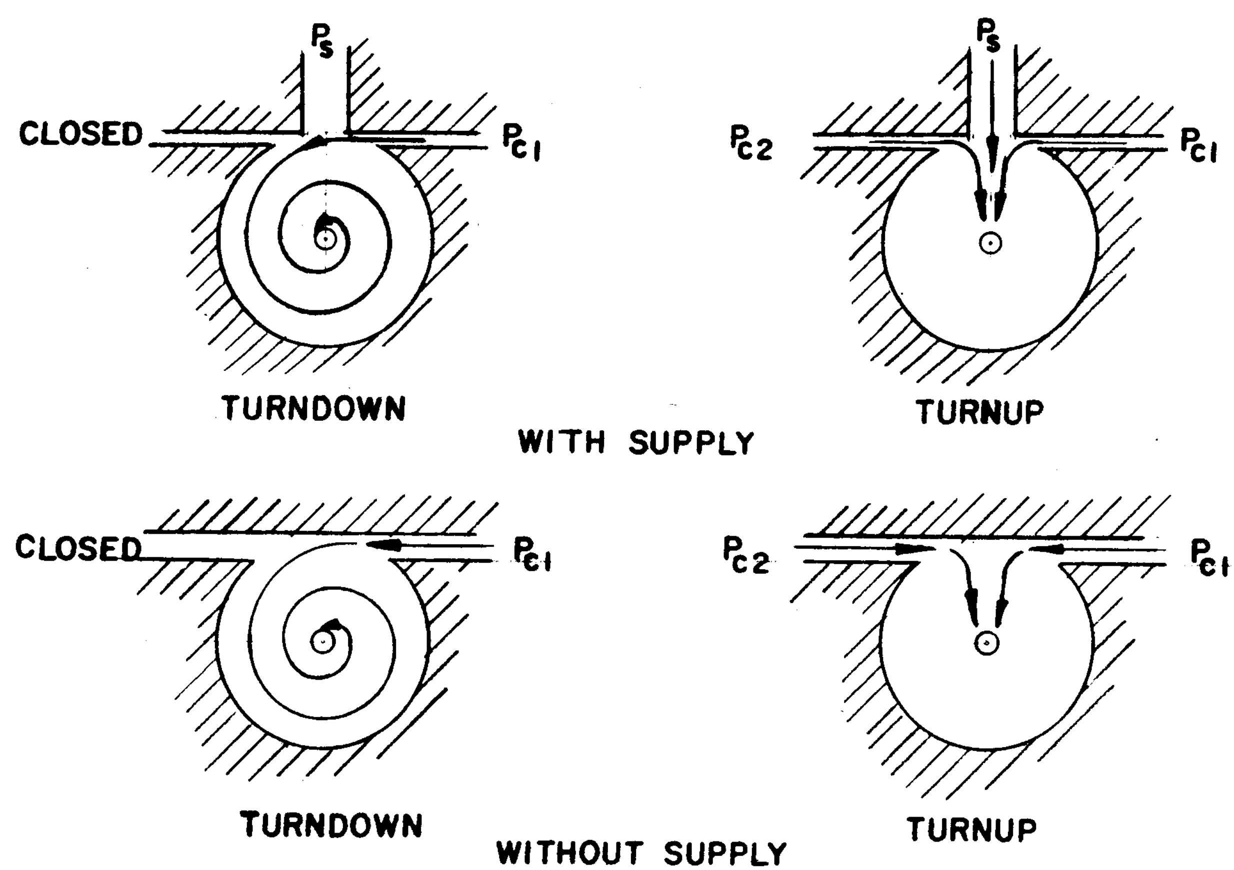


FIGURE 3-1-6 VORTEX VALVES WITH OPPOSING CONTROL PORTS

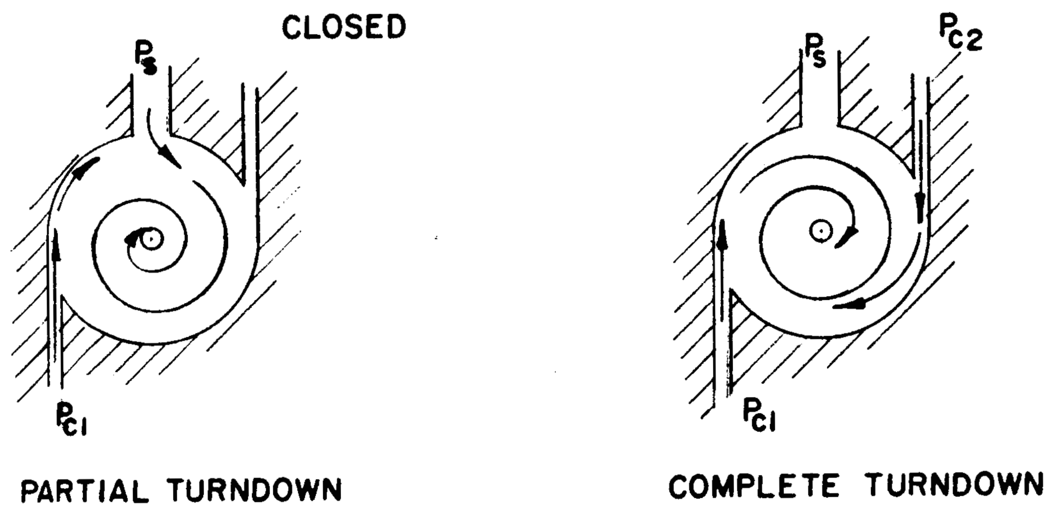
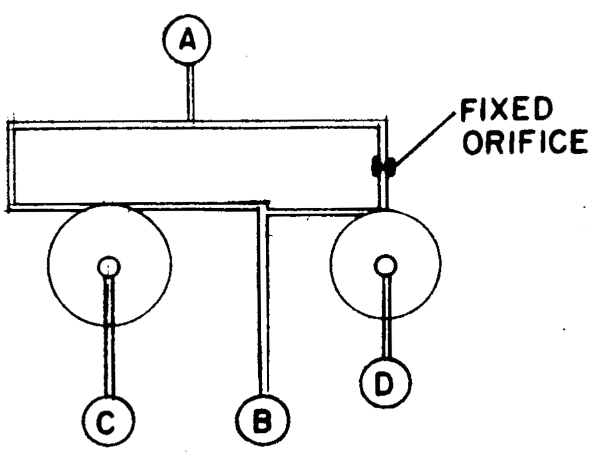


FIGURE 3-1-7 VORTEX VALVE WITH CONTROL PORTS IN SAME DIRECTION



X - DENOTES HIGH PRESSURE SIGNAL

LOGIC SEQUENCE

A	B	C	D
X	X	X	O
X	O	O	X
O	X	O	O
O	O	O	O

FIGURE 3-1-8 VORTEX VALVE - AND COMPONENT - SCHEMATIC

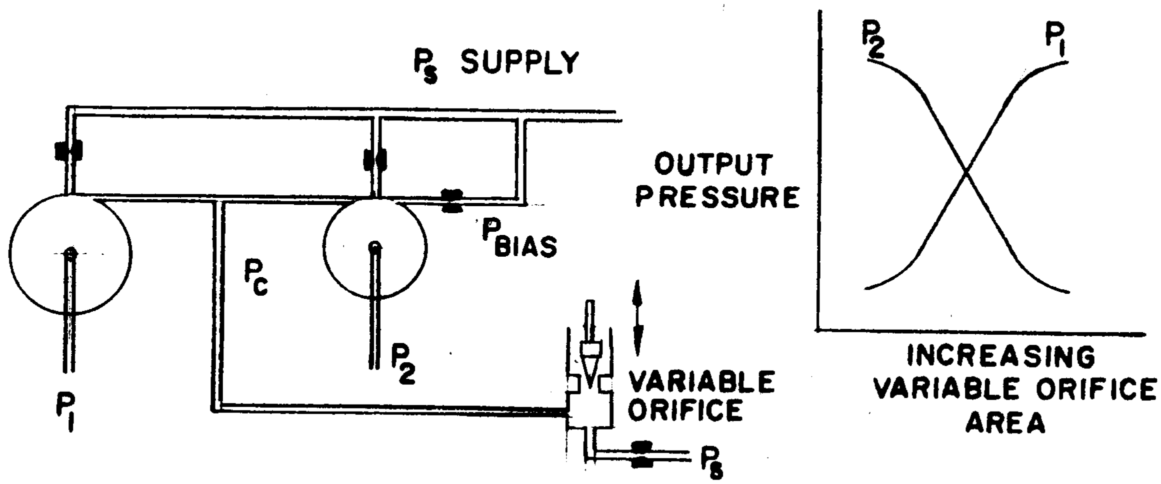


FIGURE 3-1-9 VORTEX VALVE - GATE COMPONENT - SCHEMATIC

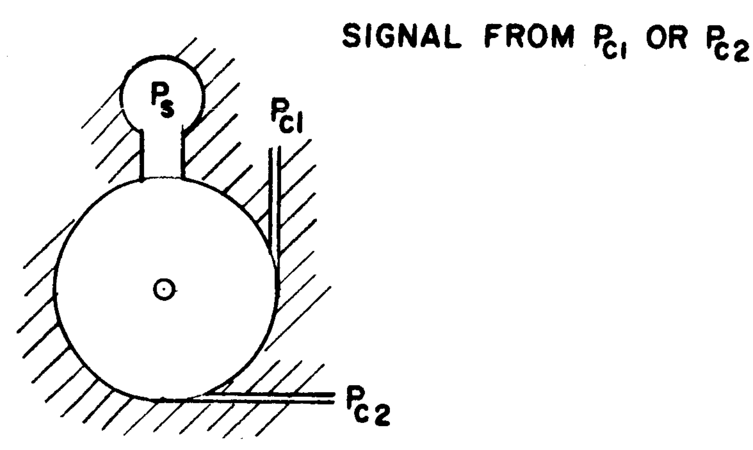


FIGURE 3-1-10 VORTEX VALVE - OR COMPONENT - SCHEMATIC
(INCORPORATED INTO POWER VALVE)

3.1.3 Vortex Valve Self Commutation Circuit

When vortex valves are incorporated into the commutation circuit to replace the AND, OR, and gate valves, the resultant commutation circuit appears as illustrated in Figure 3-1-11. The circuit is made up of thirty-three vortex valves and a directional valve. To reduce the complication of the circuit, as well as the number of vortex valves, the functions of the commutator valves and AND valves were combined. The revised vortex valve commutation circuit was evolved as shown in Figures 1-5-1 through 1-5-4. The revised commutation circuit employs eight fewer vortex valves and is considerably less complicated. The operation of this commutation circuit is described on the figures.

3.2 COMMUTATION CIRCUIT DESIGN

3.2.1 Basic Construction

Because of the environmental conditions, it was decided to fabricate the circuit of stainless steel plates. These plates contain the vortex valves and channeling as is illustrated in Figures 3-2-1(a) and 3-2-1(b). One plate contains sixteen selector valves, each .375 inch in diameter. Another plate contains the eight power valves, each .650 inch in diameter. The remainder of the plates contain all necessary channeling and reservoirs for the circuit.

3.3 RECENT TESTING

3.3.1 Optimization of Component Vortex Valves

The pressure error, selector, and power valve geometry were optimized with respect to the following parameters: (1) control area - outlet area ratio, (2) chamber diameter - outlet diameter ratio, and (3) control and supply port aspect ratios. The results of the tests conducted are illustrated in Figures 3-3-1 through 3-3-6. In optimizing each valve a compromise was made between pressure gain and turndown at one extreme, and flow gain and turndown at the other extreme. From the abovementioned curves the optimum geometry was derived.

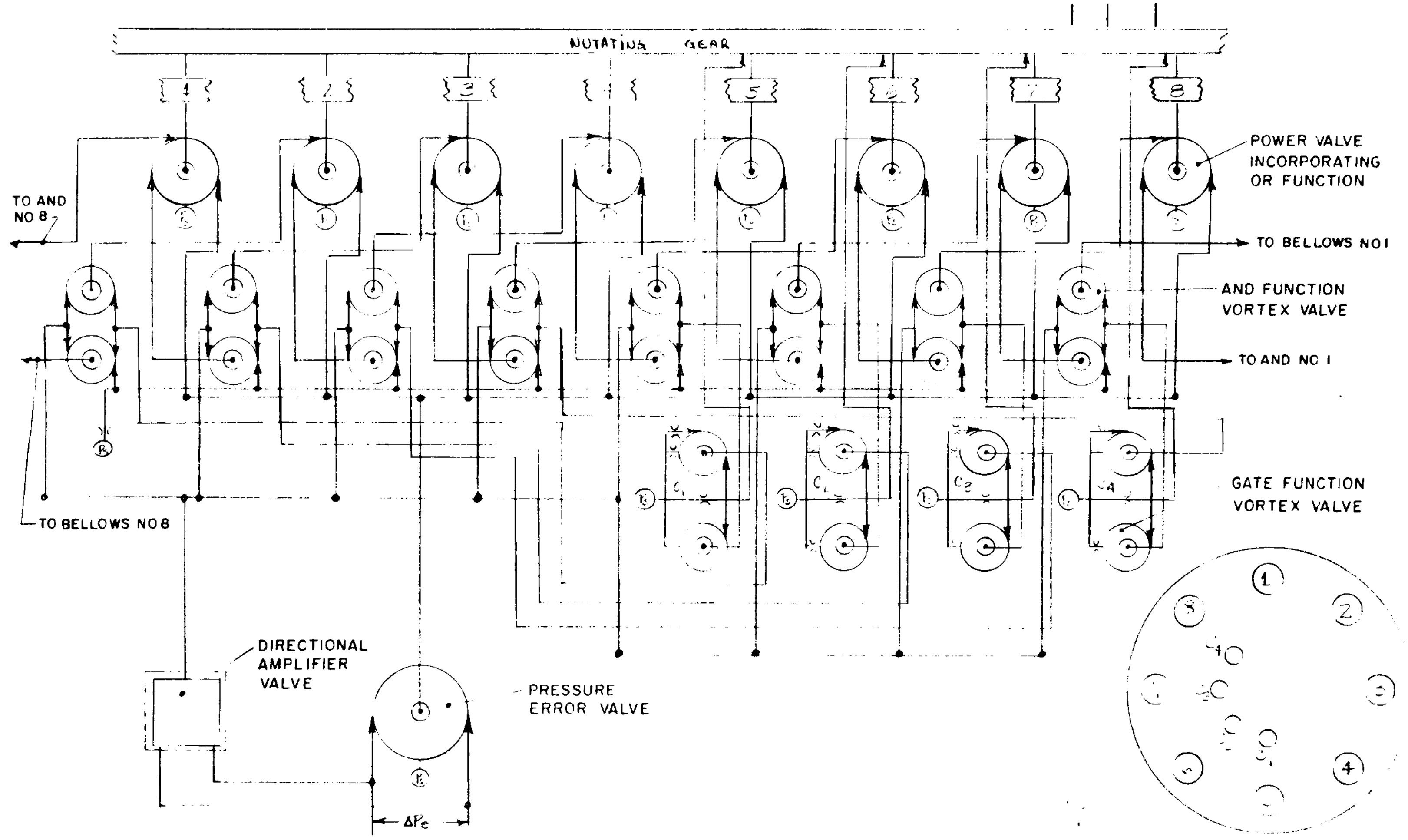


FIGURE 3-1-11 VORTEX VALVE ANALOG COMMUTATION CIRCUIT

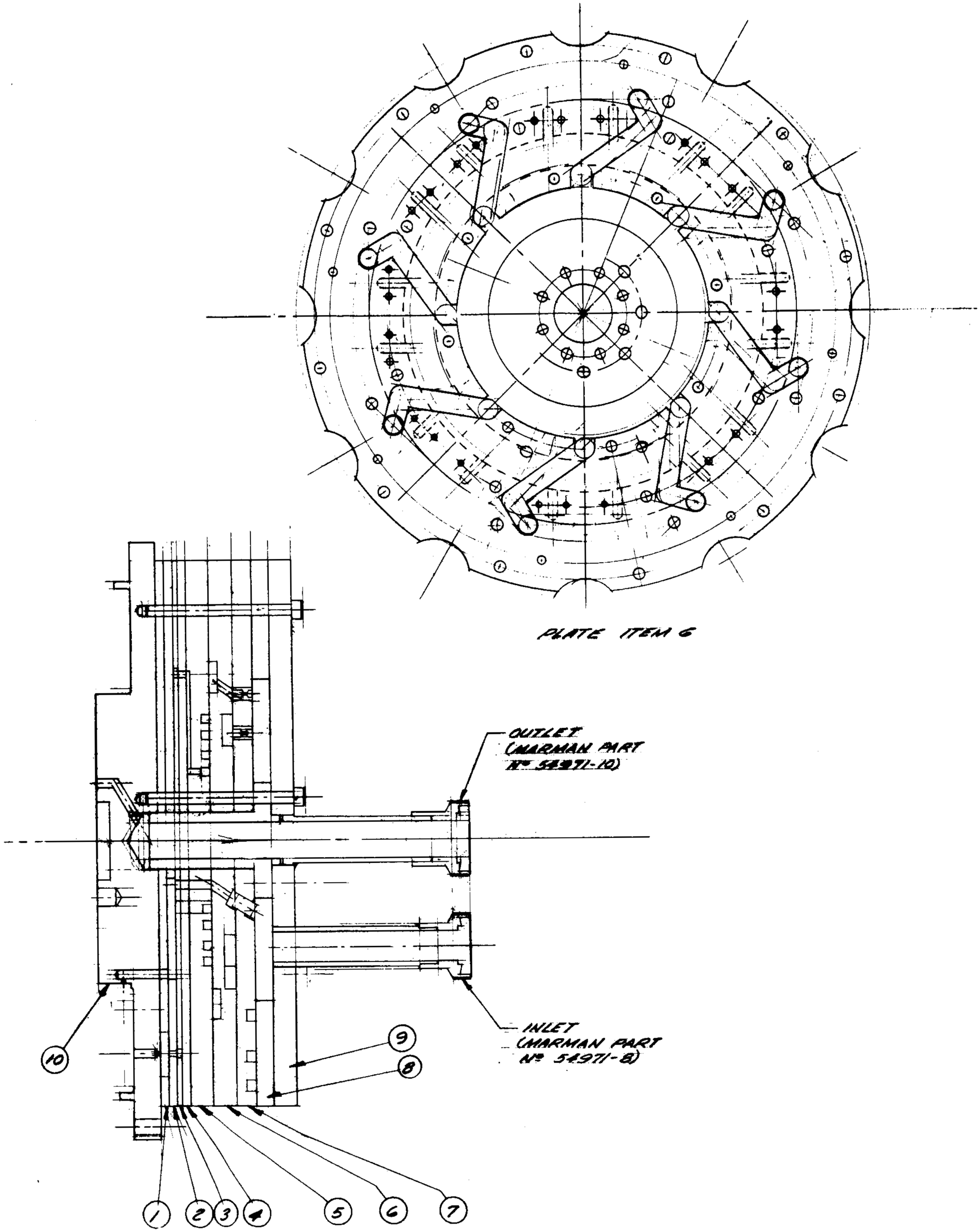
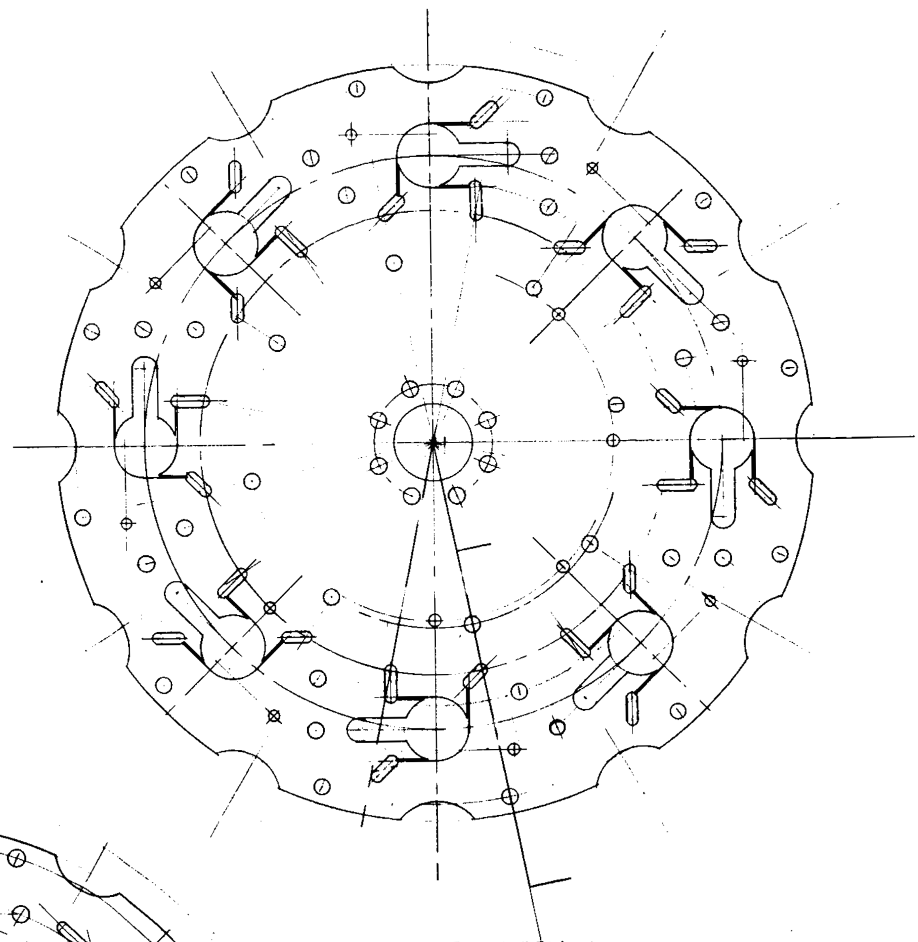


FIGURE 3-2-1(a) VORTEX VALVE COMMUTATION PLATES - TYPICAL



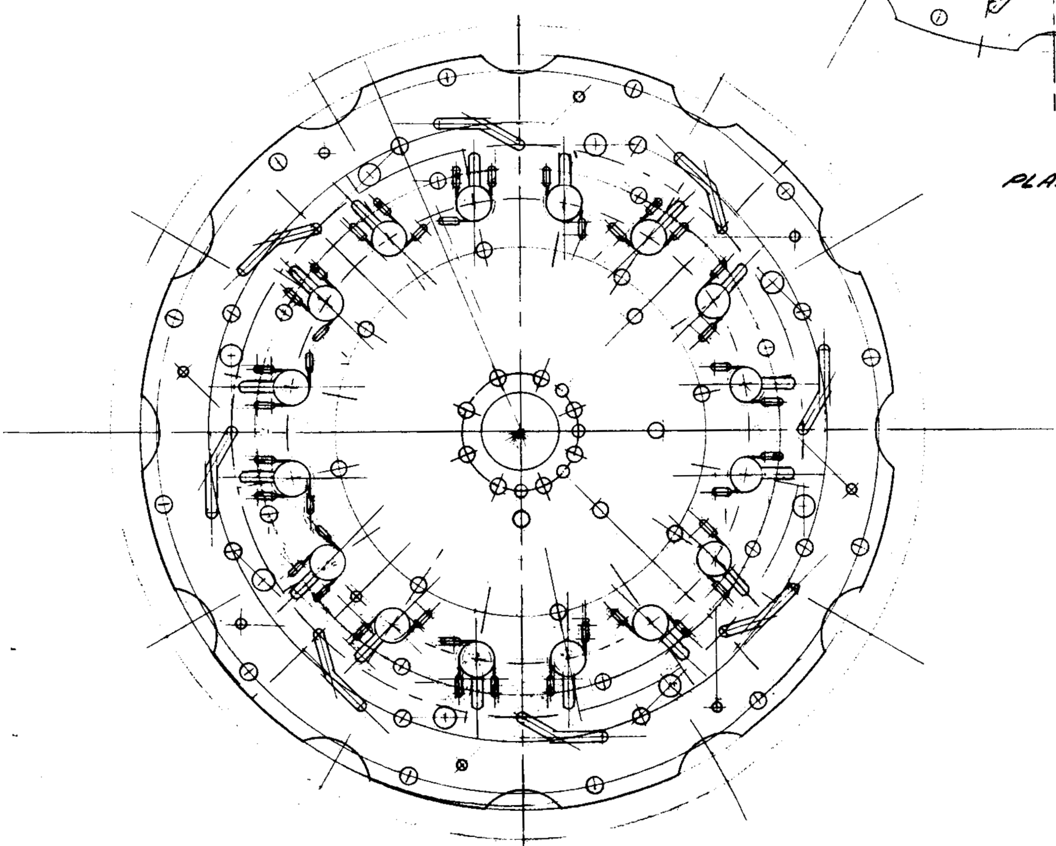


FIGURE 3-2-1(b) VORTEX VALVE COMMUTATION PLATES - TYPICAL

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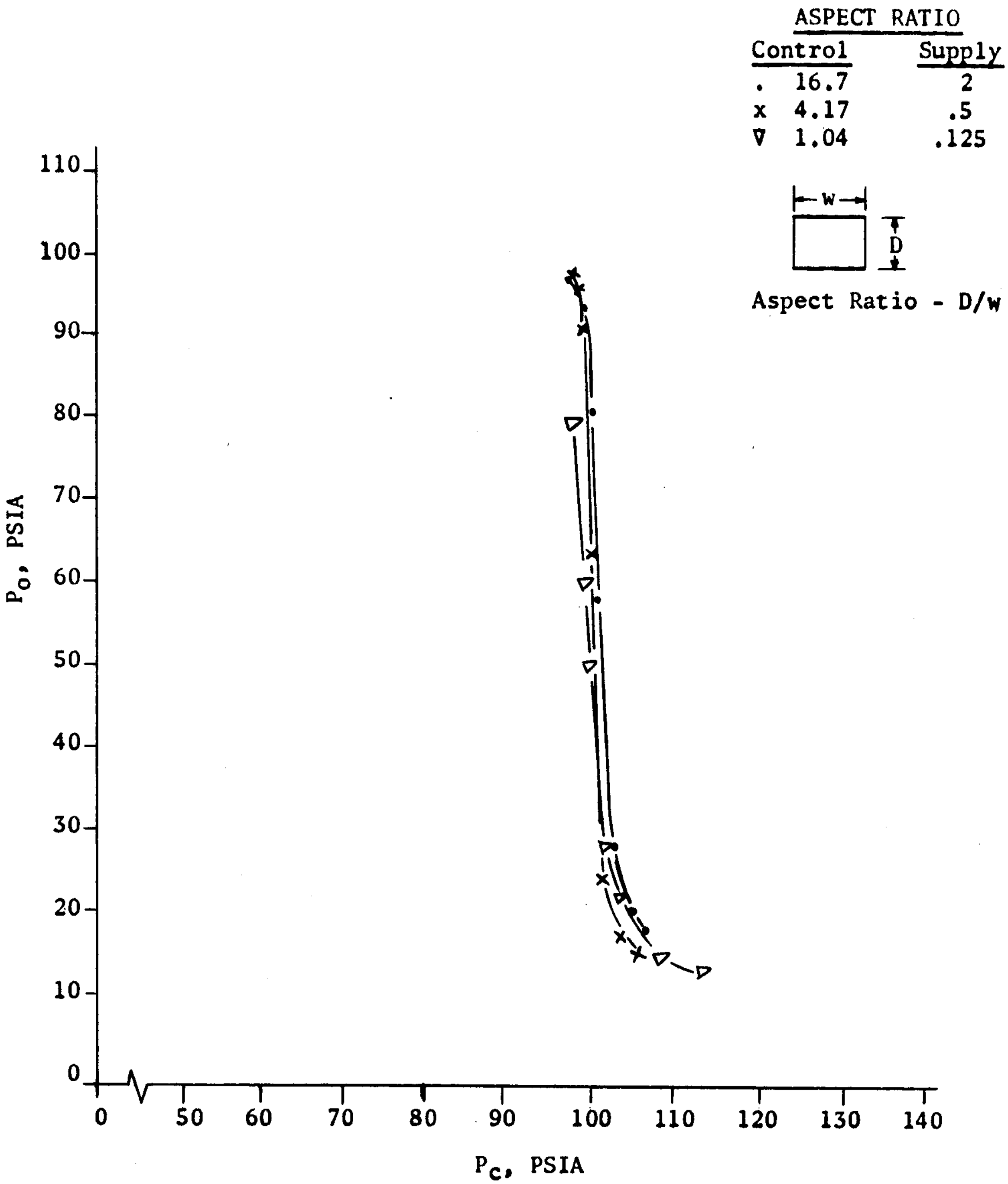
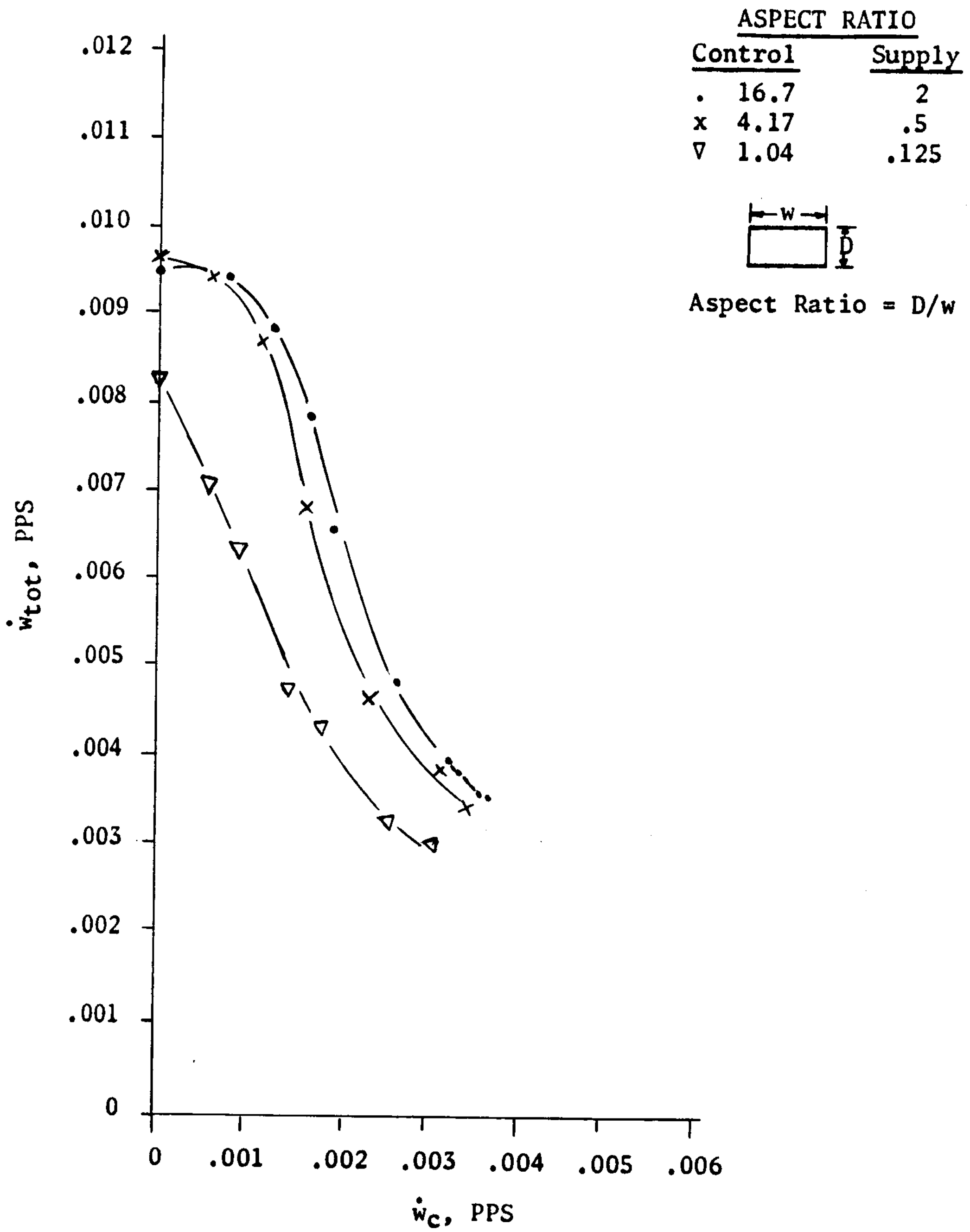


P_o - Outlet Pressure
P_c - Control Pressure

NOTE: See Page 3-24

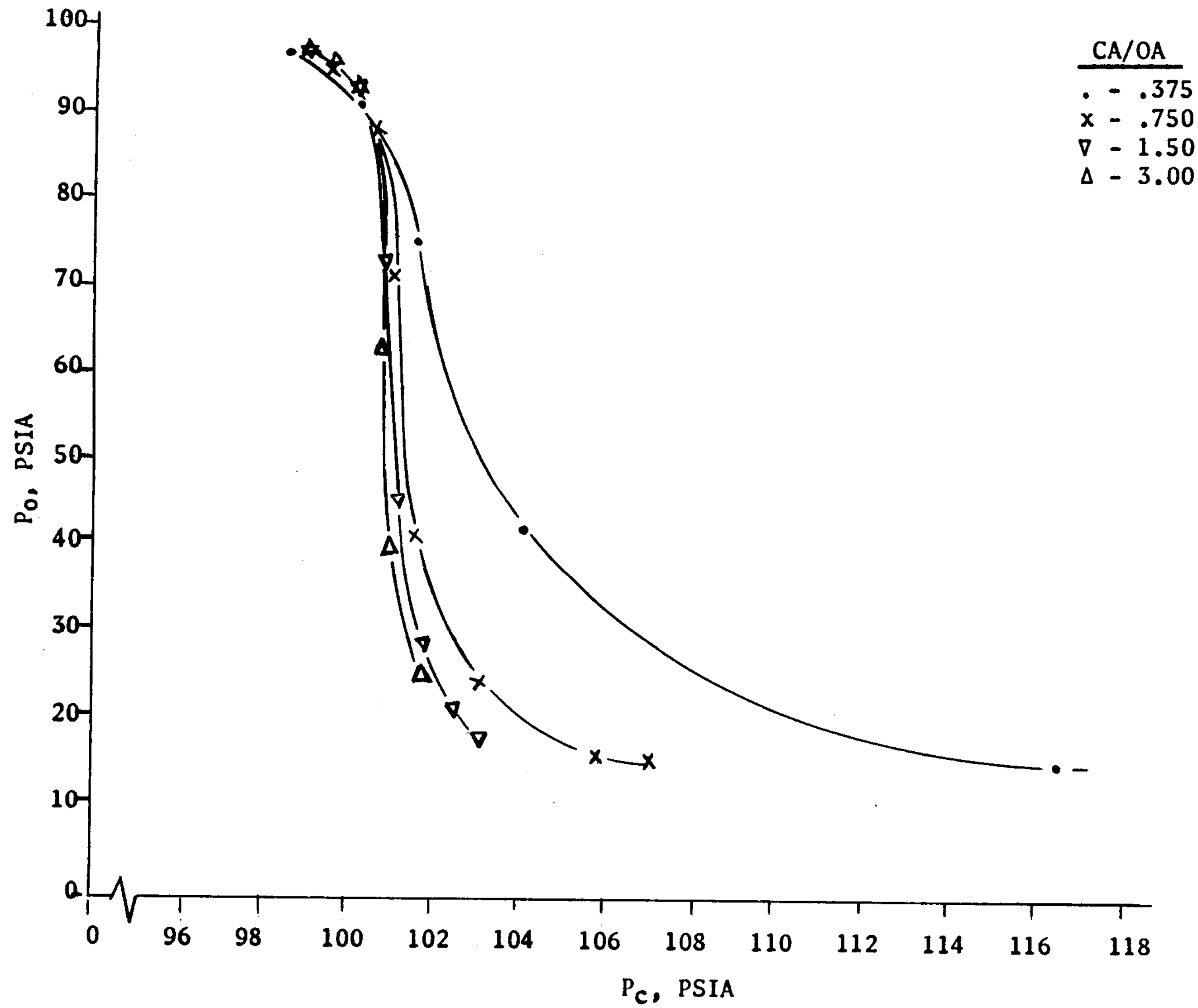
FIGURE 3-3-1 OUTLET PRESSURE VS. CONTROL PRESSURE FOR VORTEX VALVE



$\dot{w}_{tot}$ - Total Weight Flow Rate
$\dot{w}_c$ - Control Weight Flow Rate

NOTE: See Page 3-24

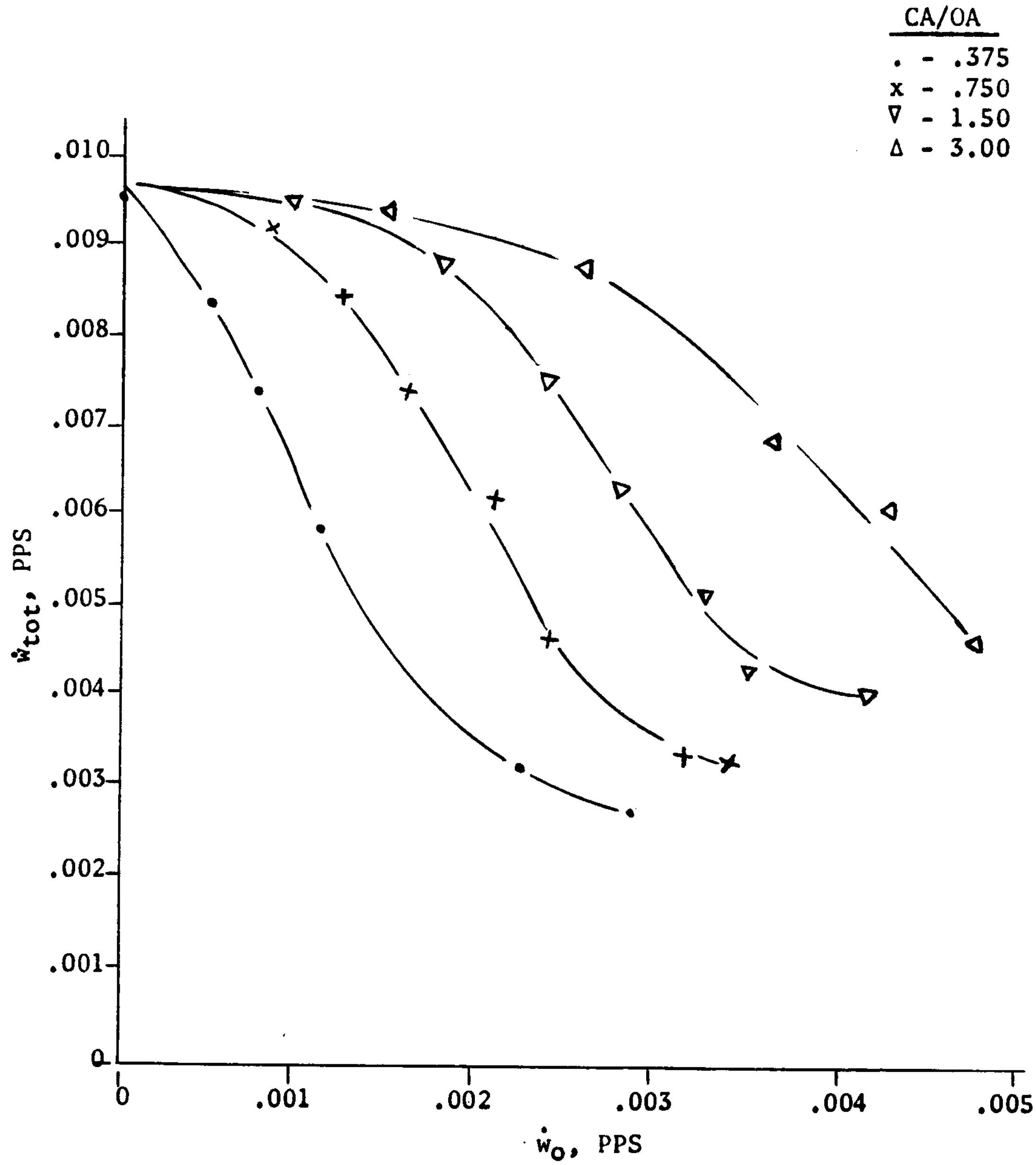
FIGURE 3-3-2 TOTAL WEIGHT FLOW RATE VS. CONTROL WEIGHT FLOW RATE FOR VORTEX VALVE



P_o - Outlet Pressure
P_c - Control Pressure

NOTE: See Page 3-24

FIGURE 3-3-3 OUTLET PRESSURE VS. CONTROL PRESSURE FOR VORTEX VALVE

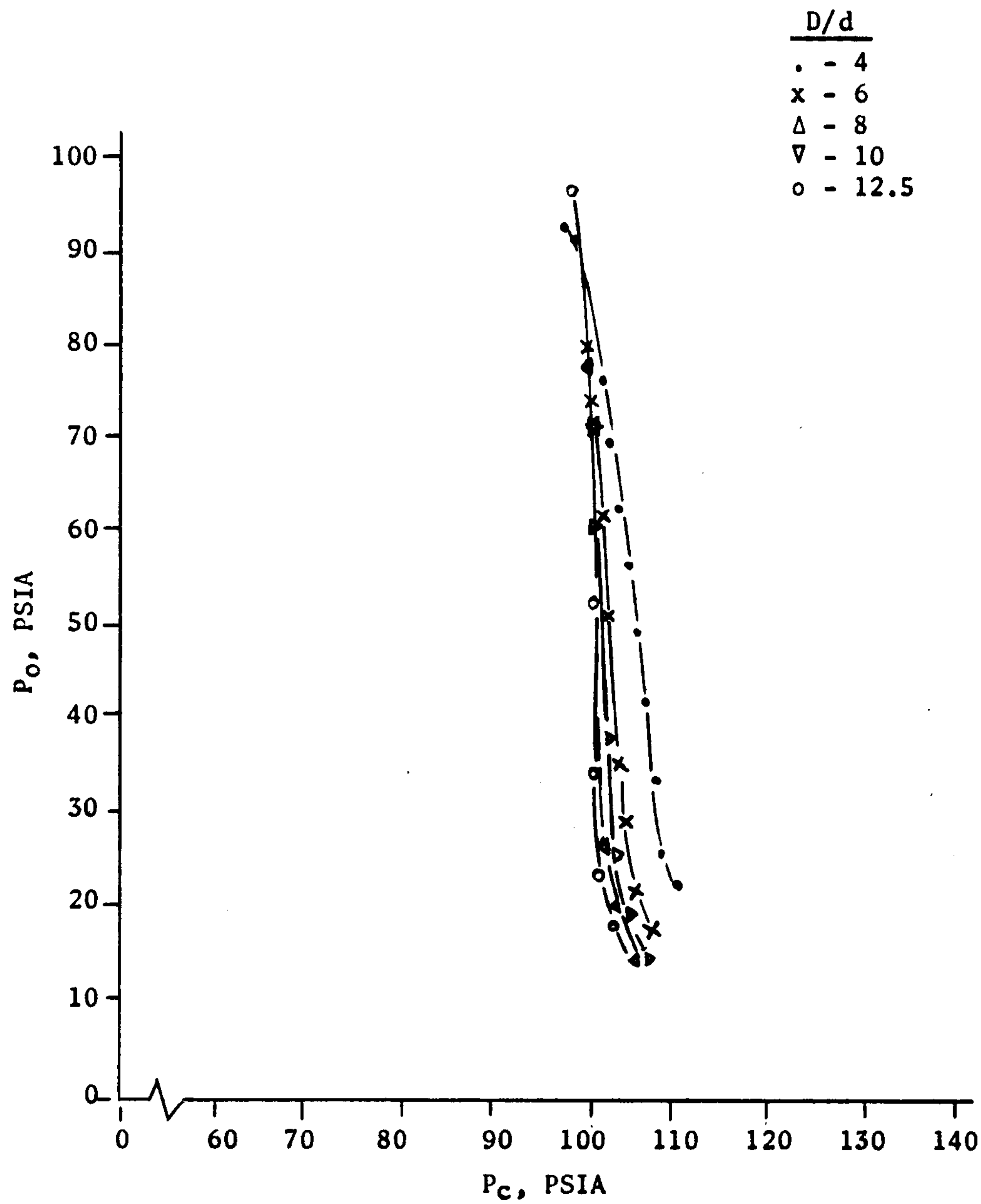


$\dot{w}_{tot}$ - Total Weight Flow Rate

$\dot{w}_c$ - Control Weight Flow Rate

NOTE: See Page 3-24

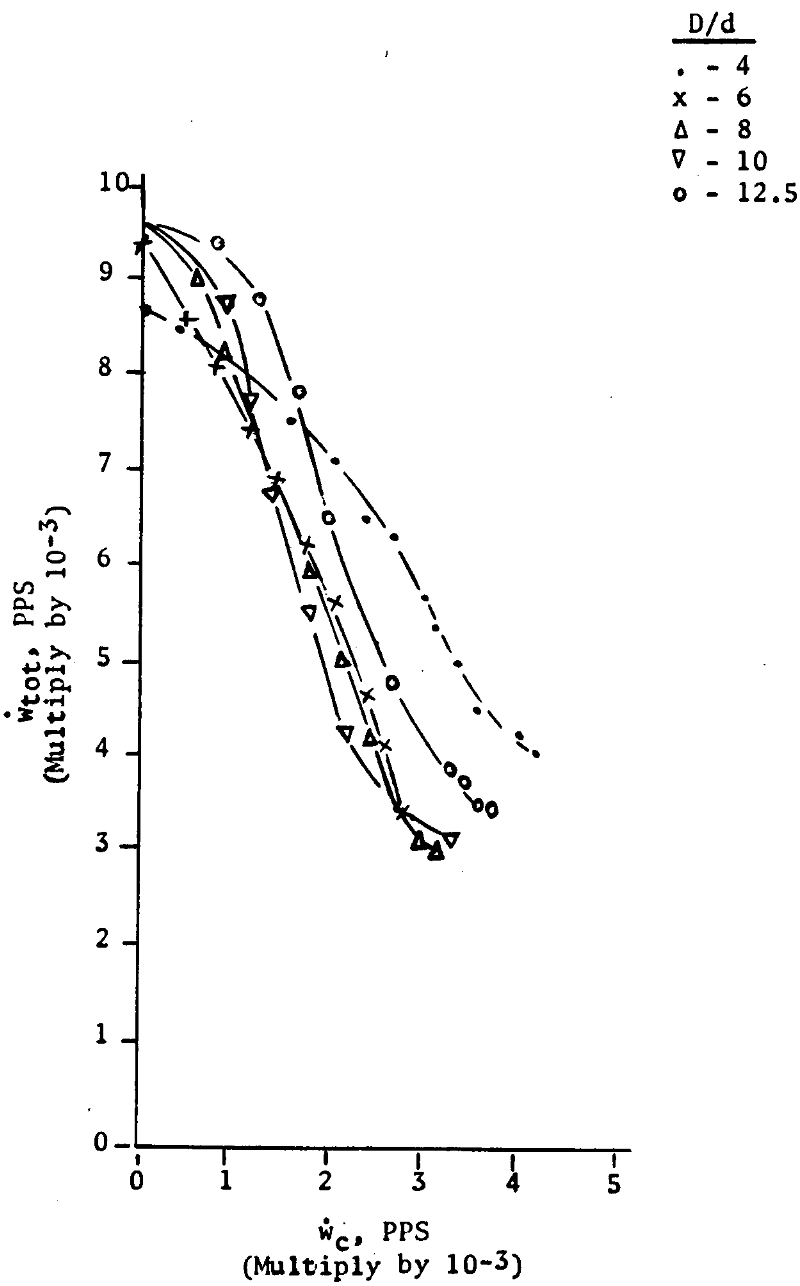
FIGURE 3-3-4 TOTAL WEIGHT FLOW RATE VS. CONTROL WEIGHT FLOW RATE FOR VORTEX VALVE



P_o - Outlet Pressure
P_c - Control Pressure

NOTE: See Page 3-24

FIGURE 3-3-5 OUTLET PRESSURE VS. CONTROL PRESSURE FOR VORTEX VALVE



$\dot{w}_{tot}$ - Total Weight Flow Rate
$\dot{w}_c$ - Control Weight Flow Rate

NOTE: See Page 3-24

FIGURE 3-3-6 TOTAL WEIGHT FLOW RATE VS. CONTROL WEIGHT FLOW RATE FOR VORTEX VALVE

3.3.2 Subassembly Tests

A power valve driven by two selector valves in parallel was simulated. The results of this test are satisfactory in that the method of obtaining the commutation logic was successfully demonstrated. Figures 3-3-7 through 3-3-9 illustrate the flow diagram for the test and the test results.

3.3.3 Plate Sealing Tests

Since the commutation circuit will consist of a number of flat plates, a series of tests were conducted to determine the best method of sealing the plates to eliminate cross-channel leakage. The test fixture used consisted of two plates bolted together at the inner and outer diameter. One plate contained four concentric slots .250 wide by .125 deep, with a radial distance between the slots of .10 inch. The second plate contained pressure tap fittings to the slots. The tests consisted of pressurizing one slot at 300 psi and observing the rate of change of pressure in the deadended remaining slots. The following tests were performed:

a. Plates machined to a 16 RMS finish and .002 in flatness. High leakage.
b. Plates hand lapped. High leakage.
c. Spray coated with "Krylon" coating. Negligible leakage.
d. Plated with .0005 inch silver, bolted together and inserted in oven for four hours at 1000°F. No leakage.

The configuration (d) was then tested at -325°F and +550°F. A temperature shock test was also performed. At no time was any leakage detected. The plates were then broken apart, and are shown in Figure 3-3-10.

From the above tests, it has been decided to use the Krylon coating for room temperature tests and adjustment. When the circuit operation is satisfactory, the silver plating procedure (d) will be used to effect a semi-permanent bond for actuator motor environmental tests.

The tests are presently being repeated to ensure consistent results.

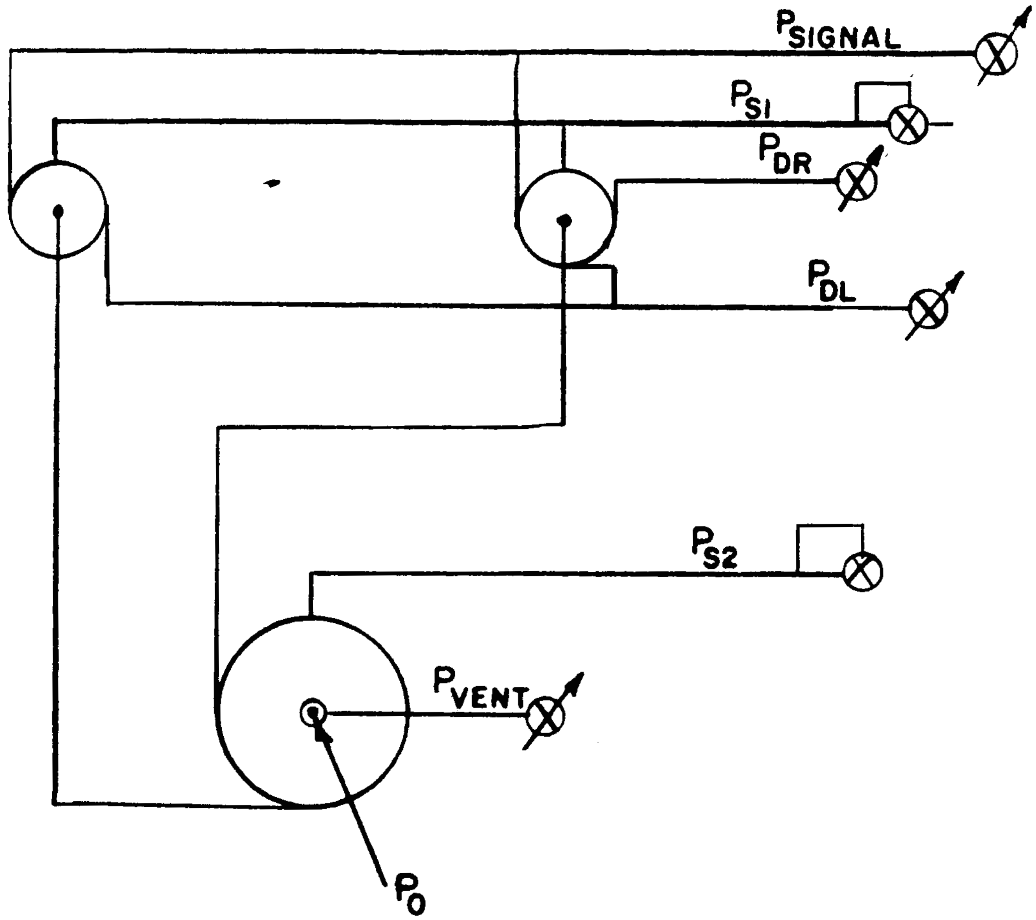
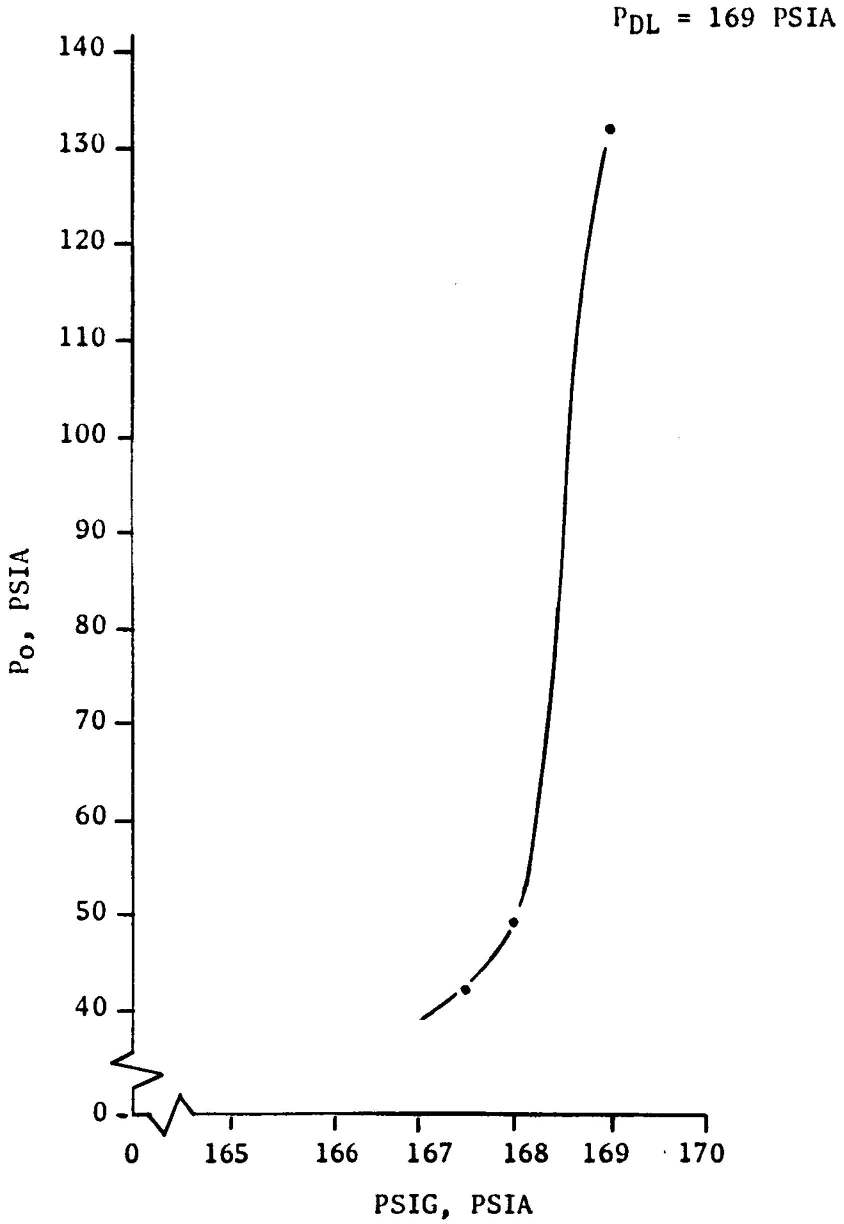


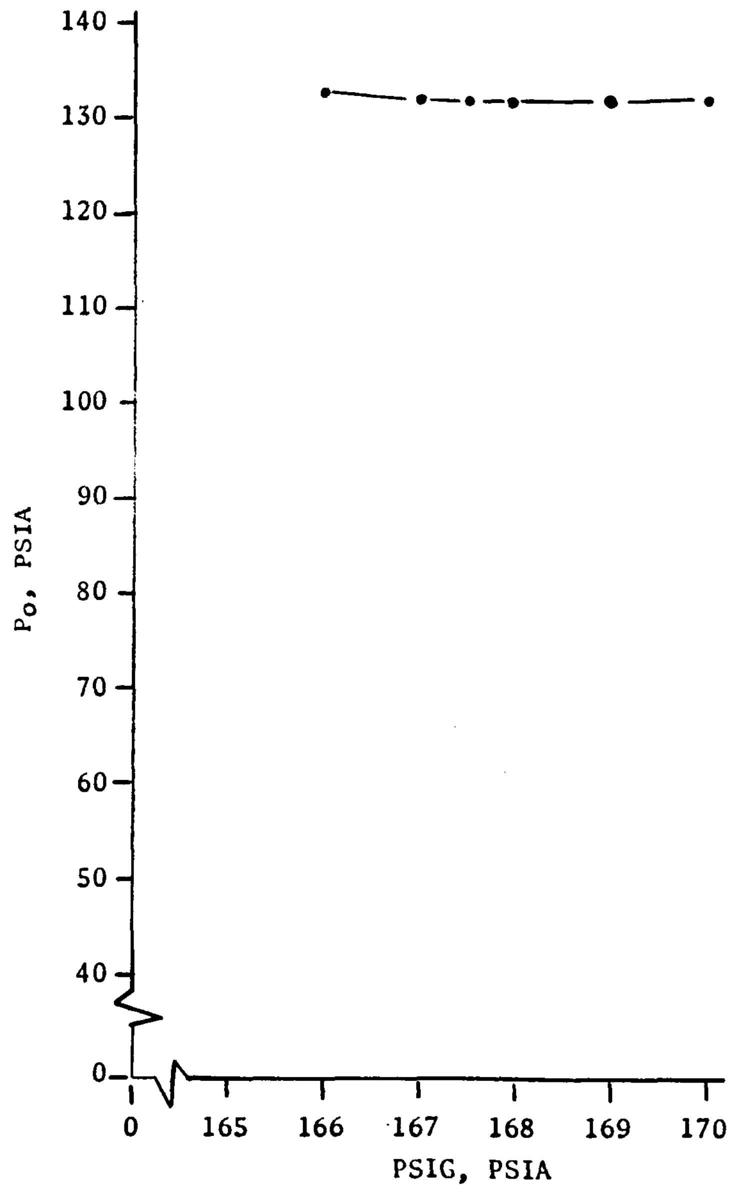
FIGURE 3-3-7 SUBASSEMBLY TEST FLOW DIAGRAM



P_O - Outlet Pressure
PSIG - Signal Pressure

FIGURE 3-3-8 OUTLET PRESSURE VS. SIGNAL PRESSURE FOR SUBASSEMBLY TEST

P_{DR} = 170 PSIA



P_o - Outlet Pressure
PSIG - Signal Pressure

FIGURE 3-3-9 OUTLET PRESSURE VS. SIGNAL PRESSURE FOR SUBASSEMBLY TEST

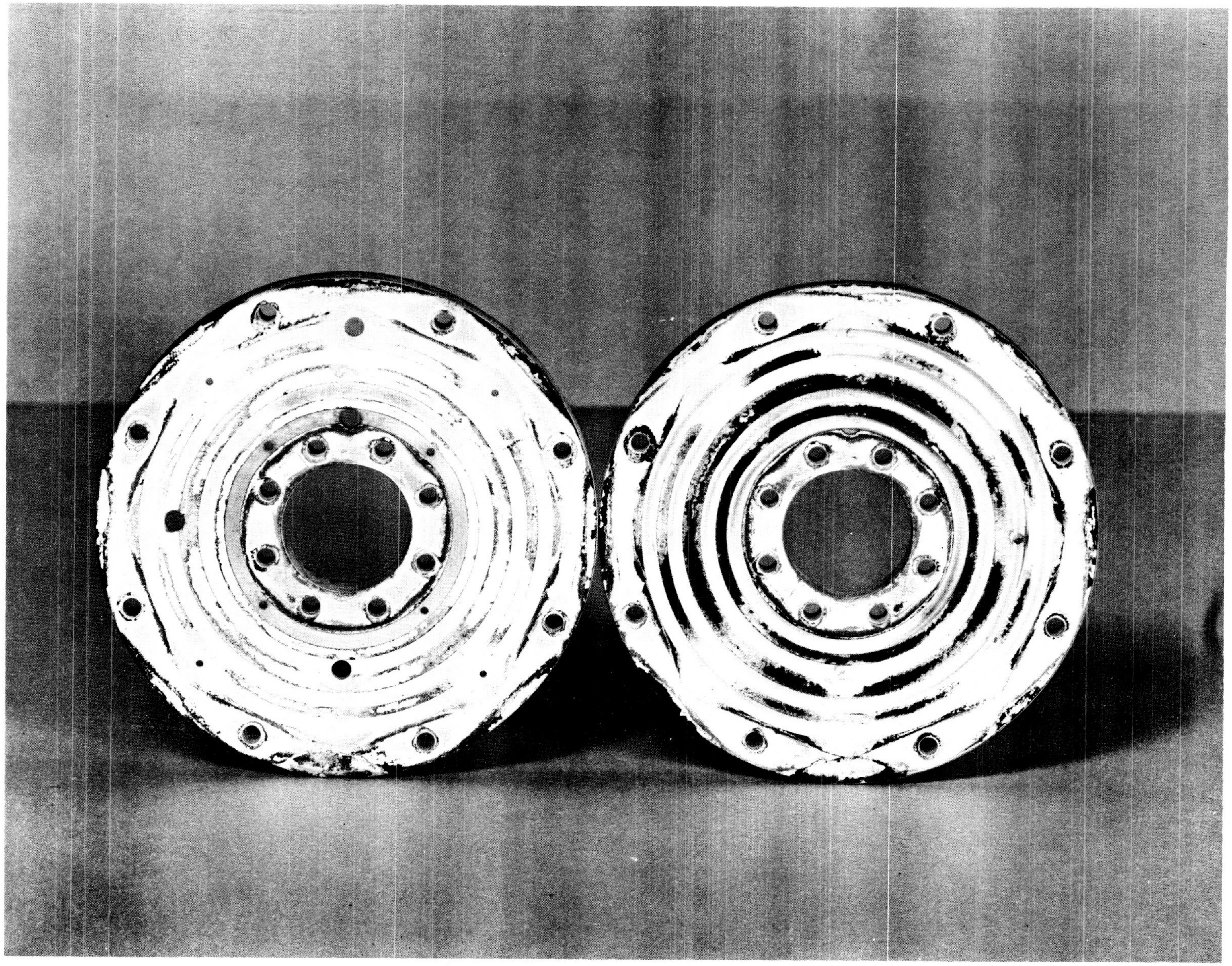
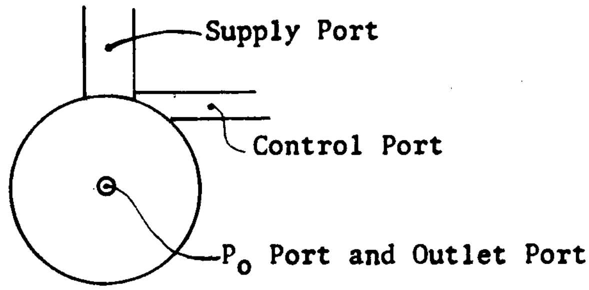


FIGURE 3-3-10 SEAL TEST FIXTURE

INFORMATION PERTAINING TO FIGURES 3-3-1 THROUGH 3-3-6



Chamber Dia. (In.)	Chamber Depth (In.)	Supply Port Width (In.)	Control Port Width (In.)	P_o Dia. (In.)	Outlet Dia. (In.)
FIGURES 3-3-1 and 3-3-2					
1.00	.25	.125	.015	.040	.080
1.00	.125	.25	.030	.040	.080
1.00	.0625	.50	.060	.040	.080
FIGURES 3-3-3 and 3-3-4					
1.00	.125	.25	.015	.040	.080
1.00	.125	.25	.030	.040	.080
1.00	.125	.25	.060	.040	.080
1.00	.125	.25	.120	.040	.080
FIGURES 3-3-5 and 3-3-6					
.320	.125	.25	.030	.040	.080
.480	.125	.25	.030	.040	.080
.640	.125	.25	.030	.040	.080
.800	.125	.25	.030	.040	.080
1.000	.125	.25	.030	.040	.080

All Vortex Valve Dimensions: ±.001 inch
Surface Finish: 16 RMS or better
All Pressure Readings: ±0.5% full scale (250 psig)
All Weight Flow Readings: ±5%

SECTION 4

SECOND QUARTER GOALS

4.1 MECHANICAL DESIGN

The mechanical components of the actuator-motor are scheduled to be completed by the end of November. Development testing will be conducted in accordance with detailed instructions issued by the Project Group. A brief outline of the testing program is as follows:

(a) Assemble and adjust gear mesh, bellows, and feedback circuit.

(b) Determine performance characteristics with mechanical commutation.

(c) Evaluate the dynamic seal.

(d) Evaluate the scram and snubbing components.

The testing will be of sufficient scope to assure the functional soundness of the actuator-motor before it is combined with the commutation circuit to form the complete actuator-motor. Minor modifications are to be made during this period to improve the performance capability of the unit. The actuator-motor is scheduled to be ready for final testing, as a complete unit, by the end of January. Detail drawings will be modified to be consistent with the hardware at the conclusion of this test period.

4.2 COMMUTATION CIRCUIT

(1) Test the model commutation circuit to determine performance characteristics of the circuit and load impedance requirements of the directional amplifier and pressure error valve.

(2) Complete drawings and fabricate the final commutation circuit. Start circuit testing.

(3) Design, fabricate and test a directional amplifier.

APPENDIX A

ENGINEERING SPECIFICATION

TORSIONAL SCRAM SPRING

PROJECT NO.	THE BENDIX CORPORATION BENDIX PRODUCTS AEROSPACE DIVISION SOUTH BEND, INDIANA	SPECIFICATION NO.	REV.
55220000		CNP-188	A

ENGINEERING SPECIFICATION

TITLE	DATE
TORSIONAL SCRAM SPRING - PNEUMATIC NUTATING ACTUATOR-MOTOR	July 1, 1964

This spring is to be used in a cryogenic-nuclear environment. Following are the specific requirements:

1.0 Environment: Hydrogen atmosphere at -350°F to +300°F

2.0 Nuclear Environment:

Component is exposed to high level nuclear radiation.

3.0 Envelope: Spring must fit unto the envelope requirements as shown on the Specification Control Drawing.

4.0 End Fittings: End fittings shall be as shown on the Specification Control Drawing.

5.0 Installed Torque: The torque supplied by the spring in the installed position shall be 50 inch-pounds (± 5%).

6.0 Deflection: The spring shall deflect 180 degrees (maximum) during operation. The torque supplied by the spring shall increase as the spring is rotated from the installed position.

7.0 Spring Rate: The spring rate shall be 4 in-lbs./rad. (maximum).

8.0 Operating Life: The minimum operating life shall be 10,000 cycles of 0-180°-0 at any temperature between -300°F and +300°F.

9.0 Weight: The weight of the complete spring shall not exceed 1.3 pounds.

10.0 Material: The spring shall be corrosion resistant and compatible with the conditions listed in the above paragraph.

PREPARED BY	CHECKED BY	APPROVED BY
J. R. Williamson	P. Every PE	J. R. Williamson JRW

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APPENDIX B

ENGINEERING SPECIFICATION

DYNAMIC SEAL

PROJECT NO.	THE BENDIX CORPORATION BENDIX PRODUCTS AEROSPACE DIVISION SOUTH BEND, INDIANA	SPECIFICATION NO.	REV.
55220000		CNP-189	A

ENGINEERING SPECIFICATION

TITLE	DATE
DYNAMIC SEAL - PNEUMATIC NUTATING ACTUATOR-MOTOR	July 1, 1964

This seal is to be used in a cryogenic-nuclear environment to seal gaseous hydrogen. Following are the specific requirements:

1.0 Medium: Gaseous hydrogen @ $-350^{\circ}F$ to $+300^{\circ}F$

2.0 Pressure Drop Across Seal: (working pressures)

Seal I.D. = P_1 = 0-650 psig
Seal O.D. = P_2 = 0-50 psig

Seal must withstand any combination of the P_1 and P_2.

3.0 Nuclear Environment:

Component is exposed to high level nuclear radiation.

4.0 Seal Envelope: As shown on Specification Control Drawing.

5.0 Lubrication Available: None

6.0 Allowable Leakage:

For ($P_1 > P_2$): 0.001 lb./sec. H_2 (0.0037 lb./sec.N_2) maximum
For ($P_2 > P_1$): 0.0001 lb./sec. H_2 (0.00037 lb./sec. N_2) maximum

7.0 Friction Torque: 15 in.-lb. maximum at any pressure drop specified in Paragraph 2.0.

8.0 Shaft Motion: Angular Velocity: 0-360 deg./sec.
Angular Displacement: 0-180 deg.

9.0 Life: 300 hours @ 100 rpm @ $-300^{\circ}F$ @ 600 psid

10.0 Mating Surface: To be specified by seal manufacturer.

PREPARED BY	CHECKED BY	APPROVED BY
J. R. Williamson	P. Every PE	J. R. Williamson WRm.

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APPENDIX C

ENGINEERING SPECIFICATION

BELLOWS ASSEMBLY

PROJECT NO.	THE BENDIX CORPORATION BENDIX PRODUCTS AEROSPACE DIVISION SOUTH BEND, INDIANA	SPECIFICATION NO.	REV.
55220000		CNP-187	A

ENGINEERING SPECIFICATION

TITLE	DATE
BELLOWS ASSEMBLY - PNEUMATIC NUTATING ACTUATOR-MOTOR	July 1, 1964

This bellows assembly is to be used in a cryogenic-nuclear environment to provide an actuating force in a pneumatic mechanism. Following are the specific requirements.

1.0 Medium: Gaseous hydrogen at $-350^{o}F$ to $+300^{o}F$.

2.0 Pressure Drop Across Bellows:

I.D. = P_1 = 0-150 psig (full range)
O.D. = P_2 = 0-50 psig (full range)
$(P_1 - P_2)$ = 70 psi (normal working pressure)

Bellows must withstand any combination of P_1 and P_2.

3.0 Nuclear Environment:

Component is exposed to high level nuclear radiation.

4.0 Effective Area: 0.435 $in.^2$ $\pm$ 5%

5.0 Bellows Assembly: End fitting dimensions per Specification Control Drawing. Bellows assembly shall contain device for reducing P_1 volume (see suggested method on Specification Control Drawing) to a minimum. If can is used, as shown, internal pressure is same as P_2 in Paragraph 2.

6.0 Bellows Motion: Stroke: $\pm$ 0.06 inch (about free length position)
Fill Time: 0.015 second (minimum)
Angular Displacement of Ends: $\pm$ $1^{o}8'$ each stroke

7.0 Life Expectancy: 50×10^6 cycles at $-300^{o}F$ ($\pm$ 0.06 inch from null = 1 cycle).

8.0 Spring Rate: 50 lbs./inch (maximum).

9.0 Material: Bellows assembly shall be corrosion resistant and compatible with conditions listed in above paragraph.

PREPARED BY	CHECKED BY	APPROVED BY
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